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TAMPERE UNIVERSITY OF TECHNOLOGY

JAVIER GONZÁLEZ MIGUEL

THERMAL SYSTEMS FOR INDOOR POOLS UTILIZING HEAT RECOVERY FROM EXHAUST AIR

Master of Science Thesis

Examiner: Professor Timo Kalema
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ABSTRACT

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The number of indoor swimming pools has increased rapidly in the last decade due to the growing demand of sport activities by the society, looking for healthy lifestyle habits. Following this growing demand, most of the Spanish cities are promoting the construction of sports centres, where all kinds of sports are practiced, including swimming.

The energy efficiency index (ODEX) in Spain has recorded a turning point in 2004 matching with the implementation of the Spanish Strategy of Energy Saving and Efficiency, which implicates more severe requirements in energy efficiency. The remarkable development of renewable energy in recent decades, has led Spain to include a requirement of minimum solar contribution in order to heat the pool water.

Solar thermal energy has undoubtedly a huge future, but need to find new applications to make it more attractive for both the end user and engineering companies. In Spain many system and facilities designed nowadays do not aim to save energy and preserve the environment, they are planned to meet the terms imposed by the new Technical Building Code. This code is applied to new buildings and rehabilitation of existing buildings for any use where there is a demand for hot water or air conditioning in indoor pools.

Along this document, a complete system to heat the pool water and condition the air in a pool enclosure in Spain will be designed. A chiller with a heat recovery system and a system that dehumidify the air from the pool with the option of recovering energy for heating purposes will be used instead a solar system as the Spanish regulations require and it will be shown that the energy input obtained from these systems is higher than the one obtained through solar panels.

PREFACE

I wrote this Master Thesis during my stay in Tampere, Finland in the academic year 2013-2014.

I would like to thank Professor Timo Kalema for giving me the opportunity of doing my Master Thesis in TUT and for his guidance and feedback.

Finally I would like to thank my family and friends, because they always support and encourage me to achieve all my goals.

Javier González Miguel

TABLE OF CONTENTS

TERMS AND DEFINITIONS

HVAC: Heating, ventilation, and air conditioning

DHW: Domestic hot water

ASHRAE: American Society of Heating, Refrigerating and Air Conditioning Engineers

RITE: Spanish regulation for thermal installations in buildings

CTE: Technical Building code

SHR: Sensible heat ratio

AHU: Air Handling Unit

HAP: Carrier’s Hourly Analysis Program

Mca: Water column meters (1 bar = 10,2 mca)

Mmca: Water column millimetres

EER: Energy Efficiency Ratio

SEER: Seasonal Energy Efficiency Ratio

F: heating effect

Q: heating energy

P: Mechanical power

$\dot{m}$: mass flow

p: pressure

cp: Specific heat

T: Temperature

hc: Convective Heat Transfer Coefficient

$\dot{V}$: Volume flow rate

ρ: Density

Ui: Heat transfer coefficient

S: Surface area

h: Enthalpy

t: Time

V: Volume

Φ_{el}: Evaporation energy loss

Φ_{rl}: Radiation energy loss

Φ_{cl}: Convection energy loss

Φ_{cond}: Conduction energy loss

σ: Stefan-Boltzman constant

W: Humidity ratio

ε: Emissivity

φ: Relative humidity

l: vaporization latent heat

1. INTRODUCTION

This project aims to study and design the facilities needed to condition the indoor air of a pool enclosure, in order to maintain its temperature, humidity and air movement at optimum levels for the users comfort. The realization of this project is led by the desire to improve the efficiency of systems in indoor pools due to the significant increase who have suffered this type of facility in Spain in recent years and the amount of energy they need.

In Spain the Technical Building Code enforces the installation of solar systems to heat partially the pool water, for this reason, that kind of systems are usually included in sport centre facilities.

That solution is usually planned to meet the terms, without seeking to save energy and preserve the environment, instead of designing a system using the existing installations destined to other purposes that could improve the energy efficiency.

Table 1.1. Distribution of buildings energy consumption in three countries [1]

Energy end uses	USA (%)	UK (%)	Spain (%)
HVAC	48	55	52
Lighting	22	17	33
Equipment (appliances)	13	5	10
DHW	4	10	–
Food preparation	1	5	–
Refrigeration	3	5	–
Others	10	4	5

As the Table 1.1shows, the HVAC plays a fundamental role in energy consumption in Spanish buildings, so special attention must be paid to everything involving those systems. Due to the huge amount of energy used by those systems, is especially interesting to find possible recovery features to take advantage of the energy released to the atmosphere.

A system to heat the pool water and condition the air in the pool enclosure based in autonomous air handling units with the possibility of dehumidification and heat recovery for pool areas, with a free cooling option using 100% outside air, will be developed. In addition it will be used a supporting system to recover the exhaust air from the chiller that is used to climatized the building.

2. BACKGROUND

2.1 Dehumidification System

2.1.1 Air Conditioning Psychrometrics

In order to understand the operation of our dehumidification system, an approach to the air conditioning psychrometrics theory will be done.

2.1.1.1 Composition of dry and moist air

According the ASHRAE's 2009 handbook, three basic definitions are used to describe air under different conditions [2]:

Atmospheric air has many gaseous components (nitrogen, carbon dioxide, oxygen …) as well as water vapour and contaminants such as dust, pollen or smoke. This is the air used for ventilation.

Dry air is atmospheric air without water vapour and contaminants. By volume, dry air contains approximately 78% nitrogen, 21% oxygen, and 1% of other gases. Dry air is used as the reference in psychrometrics (Figure 2.1).

Moist air is a combination of dry air and water vapour. The amount of water in moist air varies from zero, dry air, to a maximum defined by the temperature and pressure.

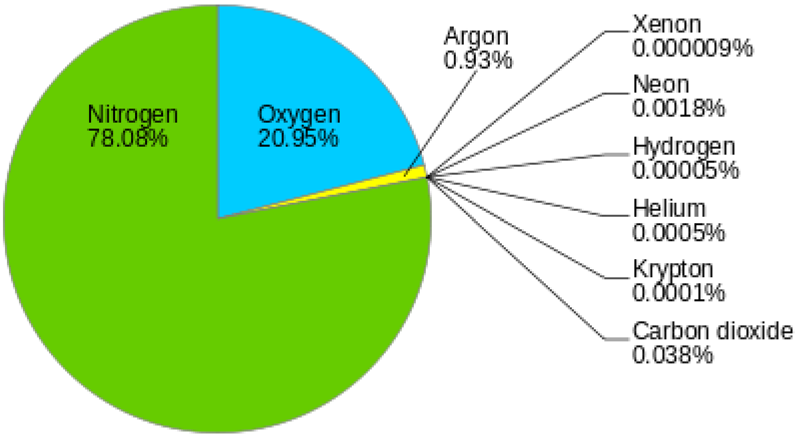

Figure 2.1. Dry Air Composition in Atmosphere

2.1.1.2 Psychrometric Chart

A psychrometric chart graphically provides the thermodynamic properties of moist air and the relationship between them, like dry bulb temperature, in vertical lines, wet bulb temperature and enthalpy, the lines sloping downward to the right, dew point temperature, the horizontal lines, and relative humidity, the curves. Two properties can be determined with any two of the others.

The ASHRAE psychrometric chart (Figure 2.2) can be used to solve mostly every process problems with moist air.

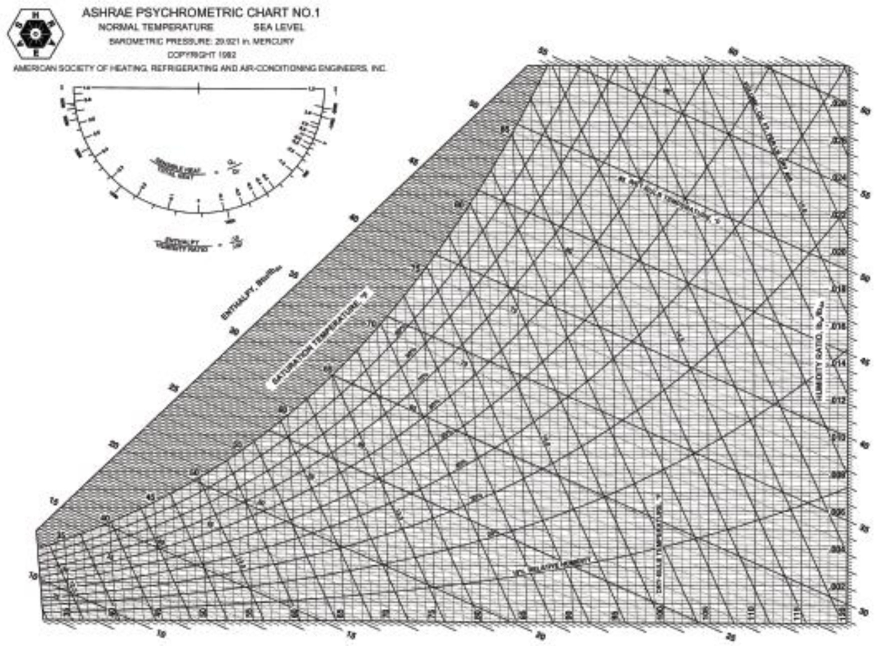

Figure 2.2.ASHRAE Psychrometric Chart

According to the ASHRAE Guide, the basic parameters relating the humidity that determine the state or condition of the air at any particular time are the ones below [2]:

- **Humidity ratio W** is defined as the relation between the mass of water vapour and the mass of dry air.

$$W = M_w / M_{da} \quad (1)$$

- **Specific humidity γ** is the fraction of the mass of water vapour in the total mass of the moist in an air sample:

$$\gamma = M_w / (M_w + M_{da}) = W/(1 + W) \quad (2)$$

- **Absolute humidity** is the fraction of the mass of water vapour in the total volume of an air sample:

$$dv = M_w / V \quad (3)$$

- **Saturation humidity ratio Ws(t, p)** is the humidity ratio of moist air saturated with respect to water at the same conditions of temperature and pressure.

- **Degree of saturation μ** is the ratio of air humidity ratio W to humidity ratio Ws of saturated moist air at the same conditions of temperature and pressure.

$$\mu = W / W_s \tag{4}$$

- **Relative humidity φ** is the ratio of the mole fraction of water vapour X_w in the air to the mole fraction of water vapour present in saturated air X_{ws} in an air sample at the same conditions of temperature and pressure. Basically, relative humidity is the ratio of the actual moisture content of the air to the maximum possible quantity of moisture that the air could have at the same temperature, so if the relative humidity is 100%, the air is completely saturated. The Figure 2.3 shows the representative lines of the relative humidity in the chart.

$$\varphi = X / X_s \tag{5}$$

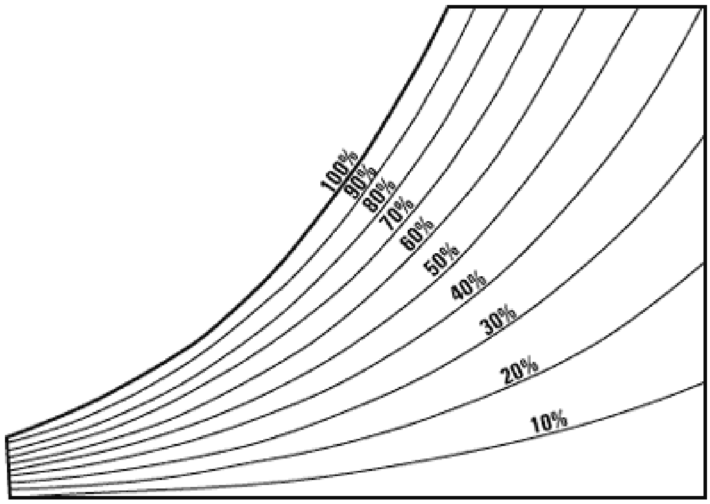

Figure 2.3. Relative Humidty Lines in the Psychrometric Chart

- **Dew-point temperature Td** is the temperature of moist air saturated at the same pressure p, with the same humidity ratio W, as that of the given sample of moist air.

- **Wet-bulb temperature T*** is a temperature related with the content of moisture in the air, the temperature is taken by covering the thermometer with a wet wick and measuring the reading as the water evaporates. Because of the cooling effect of evaporation, dry bulb temperature is always higher than dry temperature and only is the same at saturation. Wet bulb temperature on psychrometric chart is symbolized by lines with a slope from the upper right of the chart down to the lower left (Figure 2.4).

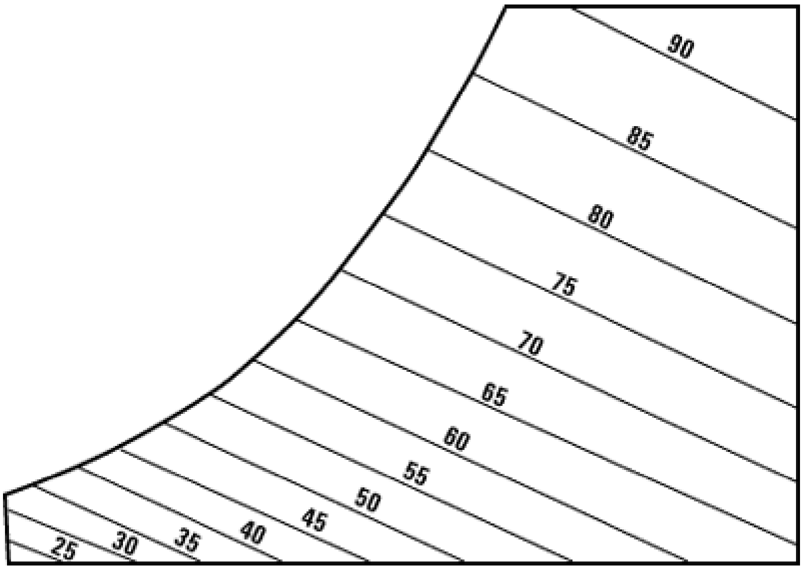

Figure 2.4.Wet-Bulb Temperature Constant Lines in the Psychrometric Chart

- **Dry Bulb Temperature** is the temperature measure with a regular thermometer. It will be considered the temperature of the air as the dry bulb temperature. In the psychrometric chart Dry-bulb temperature is found on the horizontal axis, represented by vertical chart lines (Figure 2.5).

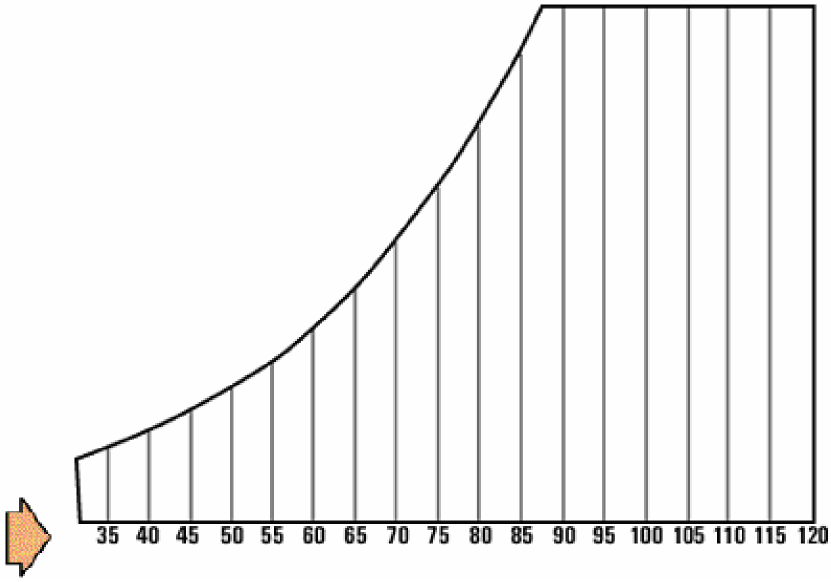

Figure 2.5.DryBulb Temperature Constant Lines in the Psychrometric Chart

2.1.2 Indoor Pool dehumidification

Humidity and moisture from an indoor pool can produce several damages if a pool dehumidifier is not sized and installed properly. Several methods can be used to dehumidify an indoor pool facility, but the variable that most greatly impacts the operating cost is the outside weather. The main goal of any dehumidification system is to keep the indoor air in the enclosure at the optimum conditions of temperature and humidity. The second goal is to achieve the previous one in the most effective way with the lowest cost. An evaluation of the effectiveness of each method and an approach to the different humidity control systems and their associated purchase, installation and operation costs will be done[3].

2.1.2.1 Push-pull ventilation

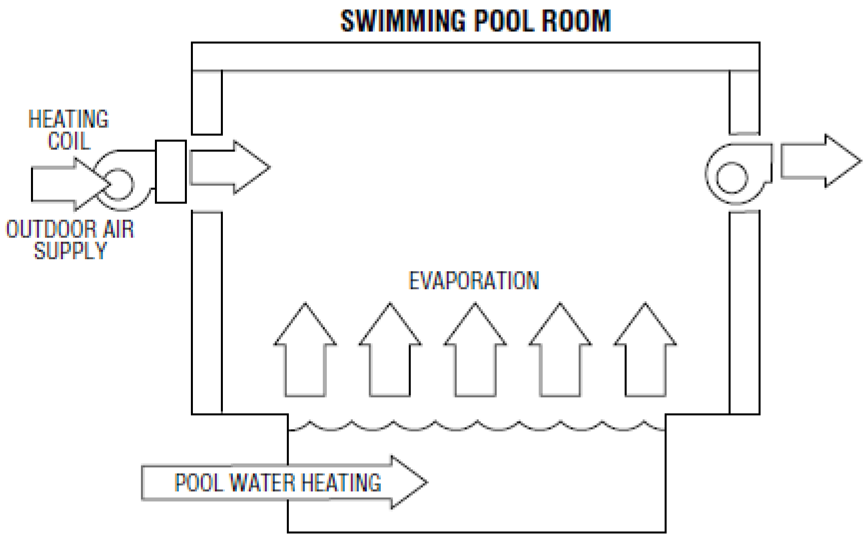

Figure 2.6. Principle of push-pull ventilation

A ventilation "push-pull" system wastes an important amount of energy by exhausting both sensible and latent heat of the enclosure introducing outdoor air (Figure 2.6). The outdoor air flow depends on the difference between the outdoor and indoor absolute humidity and it is necessary to adapt the supply air temperature with a preheating.

These systems will have similar operational costs in different climatic regions. In warmer zones, a higher outdoor air flow is required, but the cost to heat the outdoor air is smaller. In colder zones, a smaller outdoor air flow is required to dehumidify the air but significant heating costs will be needed (Table 2.1).

Table 2.1. Properties of push-pull ventilation

ADVANTAGES	DISADVANTAGES
• Low system cost to purchase • Low system cost to install	• High operation cost • Limitations in humidity and temperature control • Cooling not available on summer

In Spain the Regulation for Thermal Installations in Buildings (RITE) [4] has added a requirement of an air-to-air heat exchanger in last years, for this reason the system exposed before are no longer used in this kind of buildings.

2.1.2.2 Push-pull ventilation with air-to-air heat exchanger

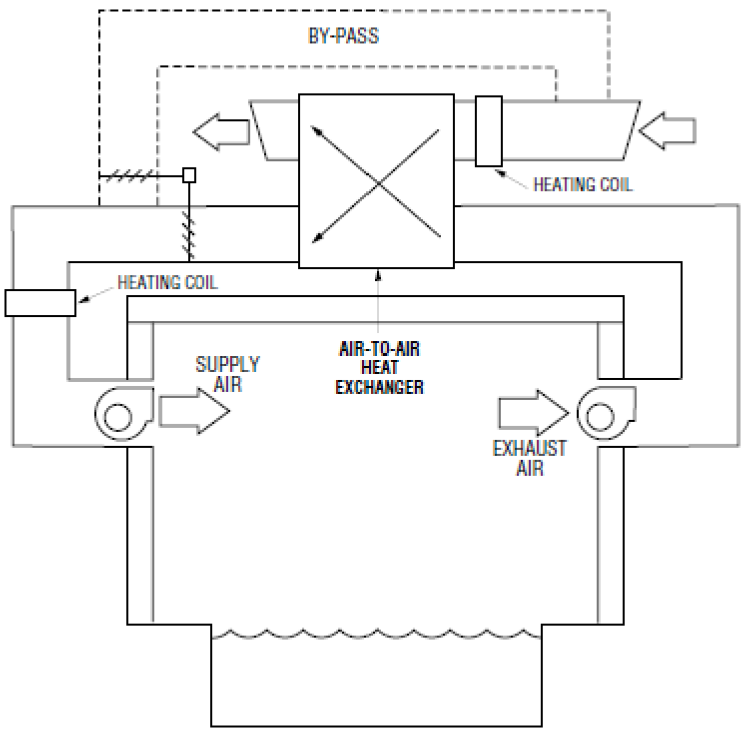

Figure 2.7. Ventilation with Heat Recovery

This method is a variation of the system studied before, including an air-to-air heat exchanger with heat recovery purposes. The incoming outdoor air flows through a series of plates and recovers the heat of exhaust air that is at a higher temperature (Figure 2.7).

The outdoor air flow has to be preheated, decreased, or somewhat bypassed when outdoor temperature is low. Efficiency is considerably reduced under low ambient environments. At cold temperatures, the need of recovery is higher and the exchanger shows the minimum recovery capacity, besides, at high outdoor temperatures cooling is not possible. When outdoor temperature and humidity are higher than the inside a condition, the heat recovery system does not perform as required (Table 2.2).

Table 2.2. Properties of a ventilation system with heat recovery

ADVANTAGES	DISADVANTAGES
• Sensible heat recovery system • Energy saving	• Low seasonal efficiency • High system cost to purchase • Does not recover all the latent heat • Limitations in humidity and temperature control

2.1.2.3 Standard dehumidifier

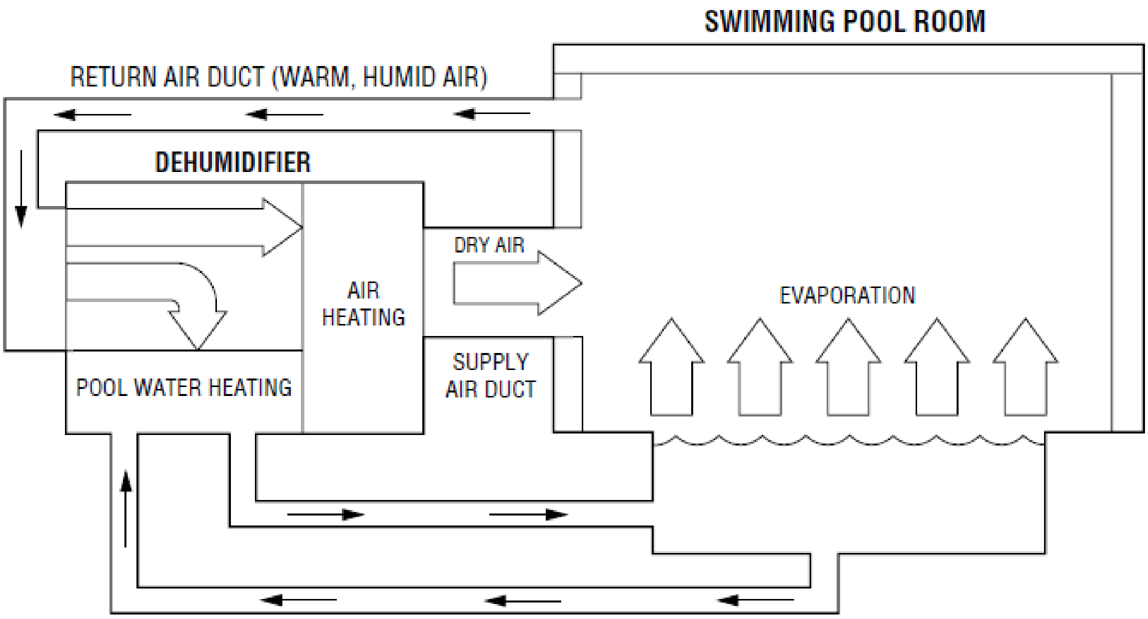

Figure 2.8. Standard dehumidifier

Three decades ago a refrigerant- based dehumidifier was designed to remove the moisture in a pool enclosure. It was the first time the outdoor conditions did not determine the system operation. The dehumidification process acted as a heat pump, turning into an energy recovery system, which returned both the sensible and latent energy back to the pool enclosure and pool water (Figure 2.8).

As the humidity is controlled by the dehumidifier, the outdoor air flow can be stopped or reduced in order to save energy.

This dehumidifier works like an air conditioner in the summer, recovering energy to heat the pool water (Table 2.3).

Table 2.3. Properties of a dehumidification system

ADVANTAGES	DISADVANTAGES
• Heat recovery system • Air conditioning • Full humidity control	• High system cost to purchase • High system cost to install

This is the basic system, but nowadays incorporates other features like free-cooling module, multi-stage filtration or support electric heaters.

2.1.2.4 Dehumidifier with "free cooling"

Through certain periods of the year, ventilation air is more effective itself than refrigeration dehumidification. This system works like an air conditioner with a saving mode, turning off the compressor when is possible to take advantage of free cooling because of the outdoor temperature (Figure 2.9).

In Spanish Regulation for Thermal Installations in Buildings [4] there is a requirement of free-cooling in systems with high powers, it will be mandatory in some circumstances.

The economizer will work only when all control points are acceptable. An economizer usually provides free cooling only 5% -20% of the year. The economizer works only a small fraction of the year, and the system essentially acts like a standard dehumidifier the remainder of the year. This dehumidifier presents a lot of advantages (Table 2.4).

Table 2.4. Properties of a free cooling dehumidifier

ADVANTAGES	DISADVANTAGES
• Full humidity control. • Energy savings by free cooling • Energy savings by heat recovery • Air conditioning	• High system cost to purchase • High system cost to install • High operational cost

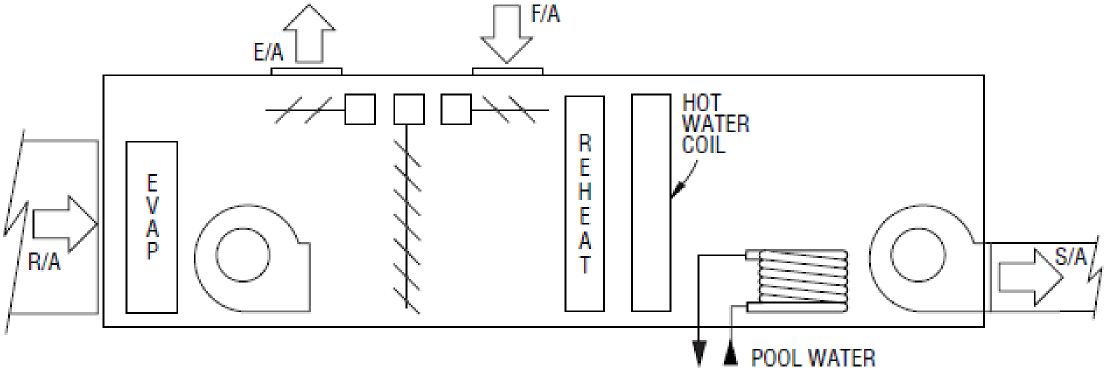

Figure 2.9. Dehumidifier with free cooling

2.1.3 Heat Loss in the Swimming Pools

2.1.3.1 Heat Loss of the pool water

The heat loss from the water and the resulting energy demands will be calculated. The main factors that determine the heat loss in the pool (Figure 2.10) are the water temperature, the temperature and humidity of the air inside the enclosure, the pool occupancy rate and the pool dimensions [5].

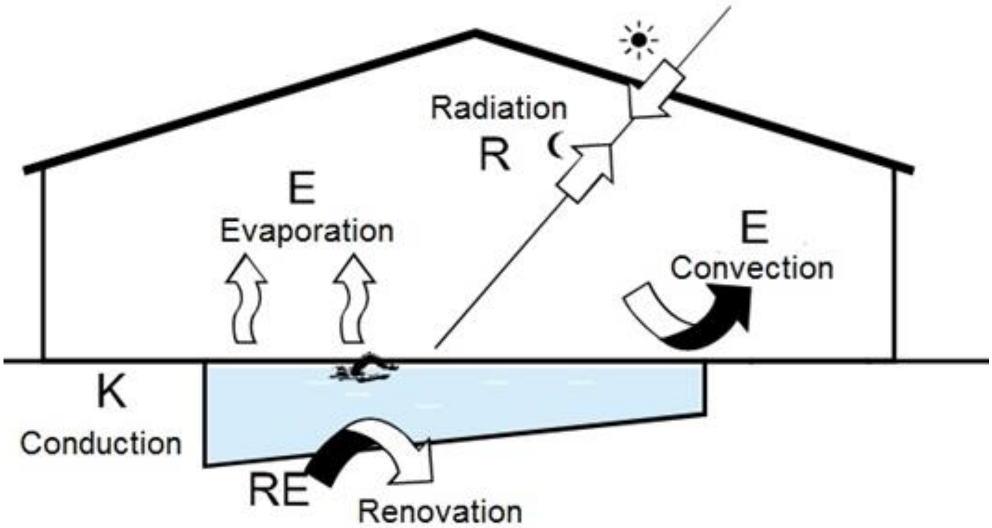

Figure 2.10. Energy balance of a pool

The total loss in the container includes three different ways of heat exchange:

Total Loss = Evaporation Loss + Radiation Loss + Conduction Loss + Renewal Loss

Evaporation loss: When the water evaporates from the surface of the pool a cooling of the rest of the pool is produced, so a decrease in the water temperature. Evaporation from the water surface will be higher, the greater the occupancy rate of the pool, and in particular the number of swimmers, because of the interaction between water and air in the turbulent flow created as a result of splash, stimulating evaporation. To calculate evaporation loss the formula below is used:

$$\Phi_{el} = \dot{m}_e \; l \; S \quad (W) \tag{6}$$

- l is the vaporization latent heat (680 $\frac{Wh}{kg}$)
- $\dot{m}_e$ is the evaporated mass flow per surface area and follows the equation proposed by Bernier:

$$\dot{m}_e = (16 + 133\, n)\,(W_w - \varphi\, W_a) + 0{,}1N \qquad (\frac{\mathrm{kg}}{\mathrm{hm^2}}) \tag{7}$$

  This expression takes into account two empirical factors: evaporation associated with the pool without water movement (coefficient 16) and evaporation associated with the splashing of the swimmers (coefficient 133n).

- n is the occupancy rate ($\frac{Swimmers}{h\,m^2}$) with the usual values 0 / 0,1 / 0,15 / 0,2 .
- W_w and W_a are the humidity ratio in saturation state, at dew point, at the water temperature and at room temperature. ($\frac{\mathrm{kg\ water}}{\mathrm{kg\ dry\ air}}$).
- φ is the relative humidity.
- S is the area from the water surface m^2.
- N is the number of spectators.

  Another equation for the evaporated mass flow considering the air velocity V could be used, as the one below proposed by Carrera, however, the difference between them is negligible [6].

$$\dot{m}_e = 9\,(W_w - W_a)\,(1+V/1{,}20)\,S + 0{,}42\,n + 0{,}08\,N \tag{8}$$

Radiation loss: To calculate the water loss due to radiation, Stefan Boltzmann formula is used. The heat exchange is a function of the difference between the average temperature of the walls and the water.

$$\Phi_{rl} = \sigma\ \varepsilon\ (T_w^{\ 4} - T_{enc}^{\ 4})\ S \qquad (W) \tag{9}$$

- σ is the Stefan-Boltzman constant (5,67 . $10^{-8}\ \frac{W}{m^2K^4}$)
- ε is the Water emissivity (0,95)
- T_w is the Water temperature (K)
- T_{enc} is the enclosure walls temperature (K)

In an indoor pool, we need to add the radiation shape factor:

$$\Phi_{rl} = \sigma\ .\varepsilon\ .\ (T_w^{\ 4} - T''_{enc}{}^{4})\ S\ F\ (W) \tag{10}$$

- T''_{enc} is the average of all the enclosure walls
- F is Radiation shape factor

Convection loss: The heat exchange between the water of the pool and the air of the enclosure due to the temperature difference between the two of them causes the following heat loss.

$$\Phi_{cl} = h_c\ (T_w - T_a)\ S\ = 0{,}6246\ (T_w - T_a)^{4/3}\ S\ \ (W) \tag{11}$$

- $h_c = 0{,}6246\ .\ (\frac{W}{m^2K^{4/3}})\ \ (Tw - Ta)^{1/3}$ [5]
- h_c is the Convective Heat Transfer Coefficient ($\frac{W}{m^2K}$)
- Tw is the water temperature (ºC)
- Ta is the air temperature (ºC)

Renewal loss: In an indoor swimming pool, there is a continuous water loss originated by evaporation, swimmers or maintenance. However, these amounts are less than the minimum renewal flow rate required by law for hygiene reasons, which varies from 1/40 - 1/50 the volume of water per day. This renewal means that heat loss will depend on the temperature of the water in the system and the temperature of the pool water and can be calculated using the following formula:

$$\Phi_{re} = \dot{V}_{RE}\ \rho\ \ Cp.\ (T_w - T_x)\ \ \ (W) \tag{12}$$

- $\dot{V}_{RE}$ is the renewal flow rate
- ρ is the density of water (1000 kg/ m^3)
- Cp is the specific heat (1,16 $\frac{J}{kg\ k}$)

- T_w is the water temperature in the pool (ºC)
- T_x is the water temperature in the system (ºC)

Conduction loss from floor: This loss depends on the floor structural characteristics surrounding the pool and the heat transfer coefficient of the material used.
Commonly a concrete floor is used and transmission loss is calculated using the formula:

$$\Phi_{cond} = \Sigma U_K S_K (T_w - T_{ext}) \quad (W) \tag{13}$$

- U_K is the Heat transfer coefficient of the floor surrounding the pool ($\frac{W}{m^2K}$)
- S_K is the heat exchange area (m^2)
- T_w is the water temperature (ºC)
- T_{ext} is the temperature outside in the surroundings of the pool container (ºC)

2.1.3.2 Heat loss in the pool enclosure

The heat transfer from the pool atmosphere to outside is divided in two ways of energy exchange, conduction through the enclosure and the energy exchange between the air from the outside and the air from the pool zone.

$$\Phi_T = \Phi_1 + \Phi_2 \tag{14}$$

Conduction loss through the enclosure: Walls, windows and other elements will be considered.

$$\Phi_1 = \Sigma\, U_i\, S_i\, (T_{in} - T_{out}) \quad (W) \tag{15}$$

- U_i is the Heat transfer coefficient of the floor surrounding the pool ($\frac{W}{m^2K}$)
- S_i is the surface area (m^2)
- T_{in} is the temperature inside the enclosure (ºC)
- T_{out} is the temperature outside the enclosure (ºC)

Energy necessary to heat the air from outside: The renewal air flow rate needed will be calculated taking into account different values for swimmers, that can be swimming or not, or visitors, that never swim.

$$\Phi_2 = \dot{V}\, \rho_E\, (h_{in} - h_{out}) \quad (W) \tag{16}$$

- $\dot{V}$ is the renewal air flow (m^3/h) = N S 22 + n S 36

    - Swimmer: 36 m^3/h
    - Visitor: 22 m^3/h

- ρ_E is the outside density air 1,2 ($\frac{kg}{m^3}$)
- h_{in} is the enthalpy of the air at the indoor temperature
- h_{out} is the enthalpy of the air at the outdoor temperature
- cp is the air specific heat (0,28 $\frac{W.h}{kg°C}$)
- n is the occupancy rate ($\frac{Swimmers}{m^2}$) and N ($\frac{Visitors}{m^2}$)
- S is the area from the water surface (m^2)

2.1.4 System start-up power requirement

In this section, the heating power to ensure the system start-up is calculated after the pool is completely filled with water. This power is calculated by the expression below, which includes the energy needed to heat the water, the pool walls and the heat loss from the water surface.

$$\Phi \text{ start-up} = \Phi \text{ water} + \Phi \text{ walls} + \Phi \text{ loss} \tag{17}$$

where the power to heat the water is obtained from the expression:

$$\text{Pwater} = \frac{V\rho C\ (Tw - Tsup)}{t} \quad (W) \tag{18}$$

- V is the volume of water (m^3)
- ρ is the water density (1000 kg/m^3)
- C is the water specific heat (1,16 $\frac{W.h}{Kg\ K}$)
- T_w is the temperature of the pool water (ºC)
- T_{sup} is the temperature of the water supply (ºC)
- t is the start-up time (usually 96 h)

The expressions to calculate the power necessary to heat the walls and the loss from the water surface were explained in the previous chapter. This loss includes the originated by radiation, convection and evaporation.

2.1.5 Dehumidification needs

Evaporation from the water surface defines the dehumidification needs. It will be greater the higher the occupancy of the pool, and in particular the number of swimmers. There are other causes that could increase the pool air humidity as the visitors or public which is an important issue in swimming competitions, because the occupation of the stands is high, or the outside air ventilation.

In accordance with the equation used before the evaporated mass flow per hour follows the equation proposed by Bernier [5]. This flow will define the dehumidification capacity:

$$\dot{m}_e = (16 + 133\ n)\ (W_w - \varphi\ W_a) + 0{,}1N \qquad \left(\frac{\mathrm{kg}}{\mathrm{hm^2}}\right) \tag{19}$$

2.2 Chiller

There are two essential HVAC systems designed to fulfill building cooling requirements, direct expansion systems and secondary refrigerant systems. In the first ones, the heat exchange between the refrigerant and the air is done directly in the units. In the second ones, chilled water is used as an intermediate medium to cool the air [7].

In water systems, one source is responsible of providing cooling throughout the whole building, and offers many reliability and efficiency advantages over individual expansion systems with a considerably lower total cost. The source of a HVAC cooling system are one or more water chillers, designed to collect the heat from the air building rejecting to the outdoor air in many cases.

The water chiller may have various configurations, with helical screw compressors, or centrifugal type and with different engine movers. The heat can be rejected to the atmosphere directly from the refrigerant to the air, or collecting the heat previously from the refrigerant with water. A water-cooled system usually offers advantages over an air-cooled system, as smaller size, longer lifecycle or better efficiency.

Vapor Compression Refrigeration Cycle

A chiller based its operation in a compression refrigeration cycle where a refrigerant evaporates from liquid to gas absorbing heat and providing a cooling effect and condensates from gas to liquid generating heat. The refrigeration cycle can be explained in four steps using the Figure 2.11:

1. Compression (1-2): In this stage, low-pressure refrigerant vapor is compressed from P1 to P2 with a parallel growth in temperature.

2. Condensation (2-3): The refrigerant at high pressure is cooled with air or water and condenses to a high pressure liquid.

3. Expansion (3-4): The high pressure liquid flows through the expansion valve decreasing the pressure and a percentage of the liquid changes to gas.

4. Evaporation (4-1): The liquid at low pressure receives heat from air or water evaporating. The low pressure vapor is led to the compressor repeating the cycle.

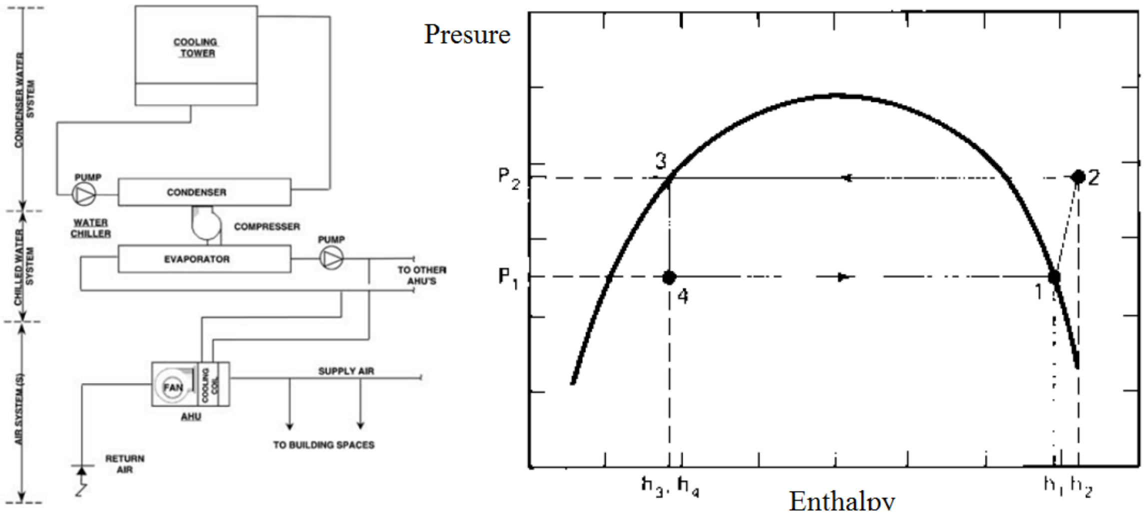

Figure 2.11. Compression cooling cycle

As shown in the figure above, each step in the refrigeration system has an associated component. The compressor raises the pressure of refrigerant vapor in the first phase, the condenser is a heat exchanger that condenses the vapor to liquid, the expansion valve controls the pressure and flow rate, from 3 to 4 points and the evaporator is a heat exchanger that causes the liquid change to gas. The Figure 2.11 represents an ideal cycle, thus in actual practice there are various inefficiencies.

The refrigerant determines the pressure-enthalpy chart and it varies with its properties. In the compression cycle, a refrigerant must fulfill some requirements: the chemically stability must be assured in the two phases, it must be inflammable and with low toxicity and the pressure and temperature ranges must be adequate for the requirements of the application.

CHILLED WATER FOR HVAC APPLICATIONS

Chilled water systems are usually used in large buildings where a central facility means less capacity, application and performance problems. The classic water-cooled HVAC system has three heat transfer loops:

The first loop is responsible for distributing the cool air by the air-handling units to the building absorbing sensible and latent heat gains originated by transmission through walls and windows, radiation through windows, infiltration and internal loads form people or electrical equipment, with the consequent increasing in the air temperature and specific humidity.

The second loop returns the air to the air handling unit, mixed with the required quantity of outdoor air, and then is led to the cooling coil where chilled water is used to extract heat from the air to redistribute it later. This process produces a water temperature increase so the water leaving will be 3-8ºC warmer than the chilled supply water. The warmer water return to the chiller where it is cooled to send it back to the AHU unit, the compressor is the element that consumes energy in the process.

The compressor produces an amount of heat that depends on the efficiency of the compressor so a last loop can be considered. This heat of compression must be taken into account to establish the amount of heat that must be rejected by the condenser to the outdoor air.

Water Supply Temperature

In order to calculate a chilled water system, firstly it is necessary to determine the required water supply temperature. The sensible and latent cooling loads must be controlled by the system regulating temperature and humidity simultaneously.

Sensible cooling is the decrease of the temperature of the air without any change in its moisture content. Sometimes the occupants, outdoor air or internal processes introduce moisture in the air causing uncomfortable atmospheres, so that to remove this excess moisture the air must be cooled below its dew point. Latent cooling is that amount of heat removed without any change in the air temperature. The total cooling load needed by a space on the water cooling coil is imposed by the sum of the previous two.

The supply air temperature required is imposed by the desired space temperature and humidity and the sensible heat ratio (SHR) calculated by dividing the sensible heat by the total heat.

2.2.1 Load calculation

Every solution to a engineering problem starts with a calculation of the duty which must be met, in this case to quantify the heating and cooling loads in the spaces to be conditioned. Modern practice is to use programs for load calculations, many load calculation software exist, with different degrees of complexity and accuracy.

In order to do an energy balance in our building the Carrier's Hourly Analysis Program (HAP) will be used [8]. HAP is a computer software which assists engineers in designing HVAC systems for buildings.

The software completes the following tasks, according the HAP user's guide:

- Calculates design cooling and heating loads for spaces, zones, and coils in the HVAC system.
- Determines required airflow rates for spaces, zones and the system.

- Sizes cooling and heating coils.
- Sizes air circulation fans.
- Sizes chillers and boilers.

The tool simulates hour-by-hour operations and generates graphics of annual, monthly, daily and hourly data as well. But only the design cooling and heating loads will be used to select the properly chiller.

A five step procedure is followed to do the calculation:

- **Problem definition.** The scope of the analysis must be determined. The type of building is involved, type of systems and equipment and other special requirements

- **Data collection.** Before calculations can be performed, some information about the building is required: its environment and the HVAC equipment to be used. The building plans will be the key to evaluate the building usage and study the HVAC system as well as determine the exposure orientations and internal loads characteristics (occupancy, lighting and electronic systems). The climate data for the building location and the materials used for all the construction elements are required

- **Introduction of the data in HAP.** The data collected has to be entered in the software:

    - Weather Data: Temperature, humidity and solar radiation are. To define weather data, the program has a weather database or the parameters can be directly introduced.

    - Space Data: A space represents a room, a group of rooms or an area with one or more air distribution terminals. To define a space, the elements must be described (Walls, windows, doors, roofs, floors, occupancy, lighting and electrical equipment). The materials used for all the construction elements are required

    - Air System Data. The equipment and controls used to cool and heat a building zone as central station air handlers, rooftop or vertical units, fan coils or water source heat pumps. The ductwork and supply terminals and controls are included.

- **Design Reports**. Once all the data has been introduced, HAP generates a system report. The report options can be selected as desired.

- **Equipment selection.** Finally, with the data from the reports, the system equipment can be selected.

HAP organizes information about the building and equipment into five categories:

- Element: Is a component of the building structure.

- Space: Is a building zone comprised of elements

- Zone: Is a group of spaces with a thermostatic control.

- Air system: Is the equipment used to provide cooling and heating to a building zone

- Building: Is the structure that contains all the systems studied.

The Hourly Analyses program is validated by many standards and regulations organizations. As the ASHRAE 90.1 - 2001 [9] and the ASHRAE 140 – 2001 [10]

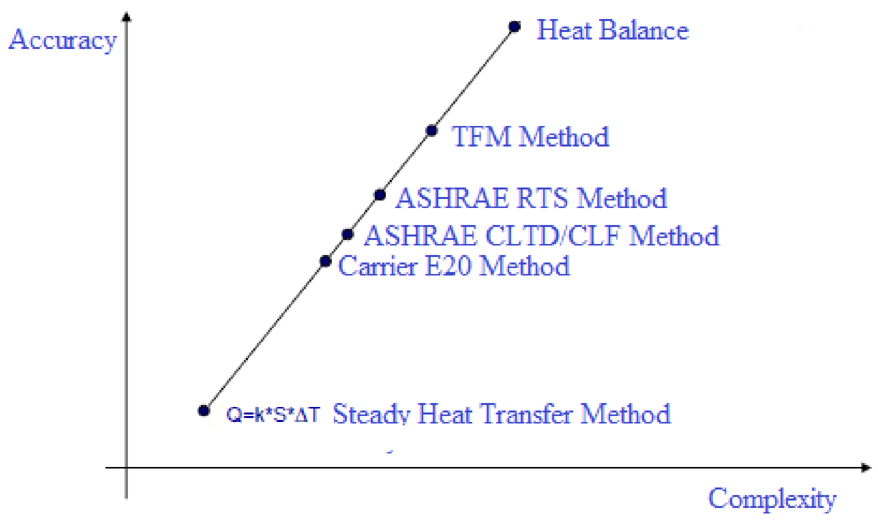

Figure 2.12. Accuracy and complexity of various thermal loads calculation methods

In the Figure 2.12 are displayed some different methods to calculate thermal loads. The Heat balance procedure is the most rigorous method for calculating thermal loads of buildings, evaluating conduction, convection, radiation and heat storage thermal processes, using the fundamental laws of Thermodynamics and Heat Transfer, but its complexity, since differential equation has to be solved, do it not suitable to resolve this kind of problem.

The Transfer Function Method (TFM) used by this software is characterized by its accuracy and flexibility. It uses mathematical algorithms simplifying the energy balance method and improving the software calculation time. This method applies a series of weighting factors, known as conduction transfer function (CTF) coefficients to the different exterior surfaces and to differences between air temperature and inside space temperature to determine heat gain with proper reflection of thermal inertia of such surfaces. Solar heat gain through glass and various forms of internal heat gain are calculated directly for the load hour of interest.

The TFM applies a second series of weighting factor, called room transfer functions (RTF), to heat gain and cooling load values from all load elements having radiant components, to account for the thermal storage effect in converting heat gain to cooling load. Both evaluation series consider data from several previous hours as well as the current hour [11].

3. CASE STUDY

3.1 Building Description

The project developed in this thesis refers to a sport centre located in Segovia, Spain. The sport centre is placed in a plot of 78,000 m² in the district of Albuera.
The building is designed for sports use, surrounded by other sport facilities as football fields, an outdoor recreational pool, athletics and tennis courts (Figure 3.1).

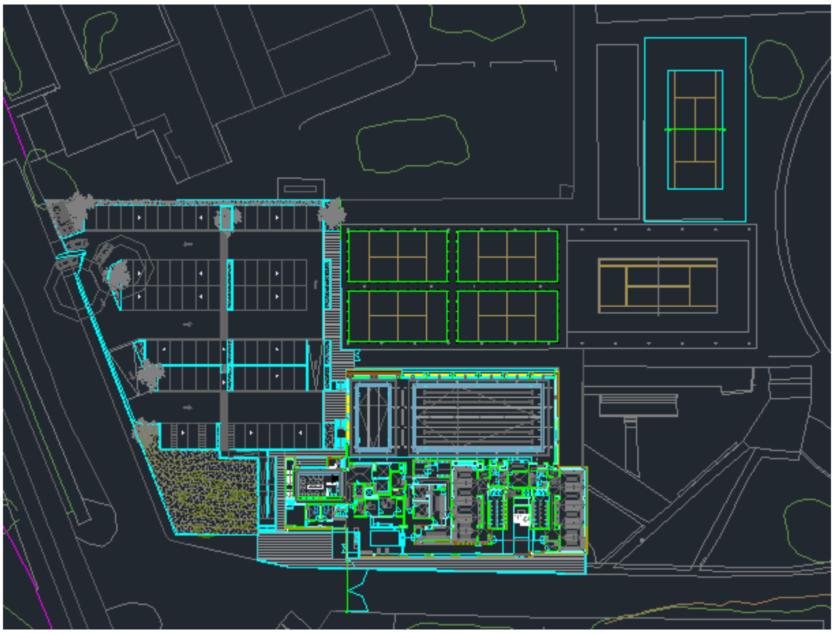

Figure 3.1. Sport centre layout

The new complex is divided in four sport areas: water sports, fitness, group sports, and paddle courts located outside. The sport centre is a three-floor building. In the ground floor, accessible from the street level, we can find the entrance, distribution corridors, information point, management office, multipurpose area and administration office. Also the water activity and swimming pool enclosure, with their own general and staff locker rooms.

In the first floor are located the rooms for individual or group physical activities.
The basement is mainly composed of rooms for maintenance and technical rooms for mechanical and electrical installations for the operation of the facilities. In this level are

placed the pumping groups, the boiler room and the electrical room. A portion of the space is occupied by the pool walls.

In the roof the remaining facilities as chillers, AHU-units, other pumping units and electric generator are located.

The area of each floor is presented below:

First Floor	1.072 m²
Ground Floor	1.825 m²
Basement	1.205 m²
TOTAL	4.102 m²

The facilities covered by this project are those needed to heat the pool water and condition the air in the pool enclosure. A chiller with a heat recovery system and dehumidification system with the option of recovering energy for heating purposes will be used. The design of the building facilities respond to sustainability criteria, energy optimization, security, usability and maintenance.

3.2 System Description

A system to heat the pool water and condition the air in the pool enclosure based in autonomous air handling units with the possibility of dehumidification and heat recovery for pool areas, with a free cooling option using 100% outside air, will be developed. Specifically, two units, one for a swimming pool area and another for a pool intended to water activities. Those units dehumidify the air and heat partially the pool water recovering energy from the heat produced in the condensing coil. The performance of that equipment will be explained in following chapters.

In extreme weather conditions it is hardly impossible to heat the water of the pool only with those units, because the heat demanded by the air is too high, which leads to consider auxiliary systems.

The Figure 3.2 represents a diagram consisting of a dehumidifier with an energy recovery system and free-cooling and a solar collectors system to heat the pool water and produce domestic hot water. The heating sources for the pool water are the boiler, the solar collectors and the dehumidifier recovery system.

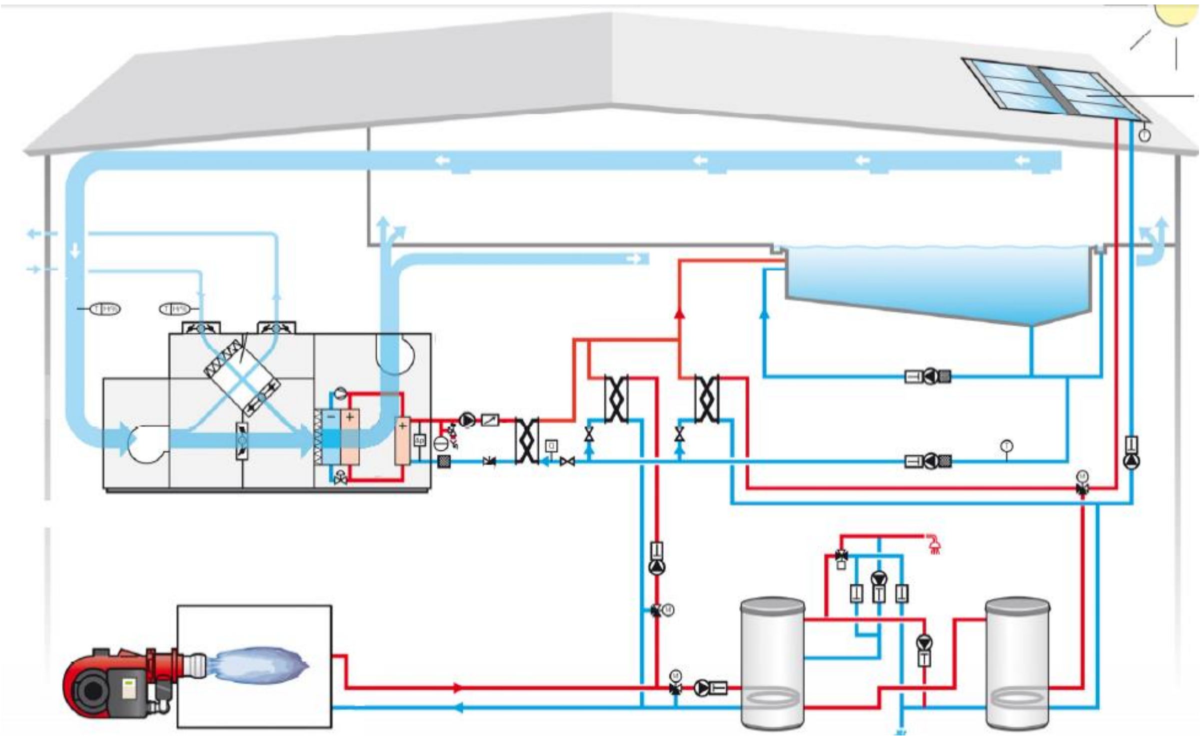

Figure 3.2. Pool heating system with solar collectors

The installation of solar technology is mandatory by Spanish regulations to cover the 50% of the hot water pool demand, but if it is proved a more efficient way to cover that demand, it is possible to avoid including such an expensive system.

This thesis seeks to find a more efficient means of generating hot water than with solar energy, through the application of air-cooled chiller systems with heat reclaim capabilities to reduce the energy consumption in the sport center.

The sport center HVAC system will need a chiller and a boiler to cool and heat the water used in the AHU-units and fancoils. To take advantage of those systems it will be developed a method to recover the heat of the chiller that normally is rejected to the atmosphere and replace the solar installation. The system proposed in the Figure 3.3 is similar to the previous one without considering the domestic water heating and introducing the recovery system of a chiller. The boiler is connected with other disposition because it will be only used to heat the water the first time or as chiller support if there were any failure. In addition the boiler will supply the HVAC system.

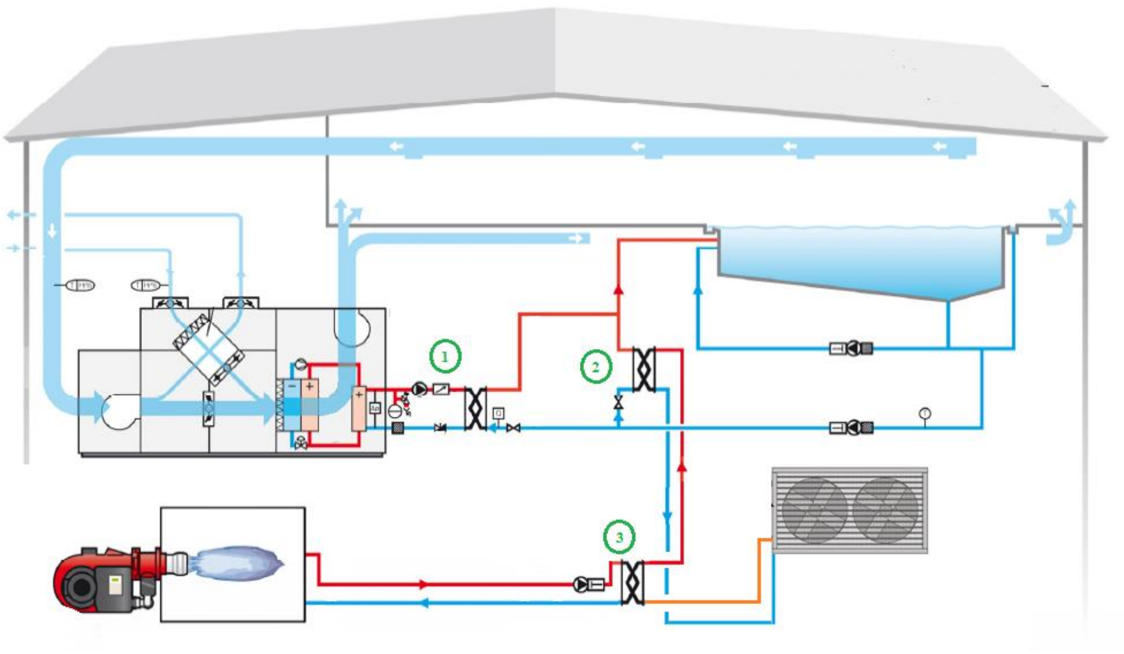

Figure 3.3. Pool heating system with a chiller recovery system

The water heat source will be as follow:

1. The dehumidifier water condensing coil.
2. The chiller heat recovery system.
3. The boiler.

The pool air heat source will be the dehumidifier water condensing coil. Moreover, it is possible to add a feature to have an extra water coil connected to the boiler in case more heat is needed, but our dehumidifier will have power enough to heat the air in the worst conditions.

In the Figure 3.4 it is showed the design of the system and the hydraulic network in primary-secondary configuration is, the primary is connected to the supply units and the secondary is responsible to distribute the water to the consumption or intake units.

For cold water, there are two secondary loops, one for AHU units and other for the fan-coil units. The primary loop is designed for a temperature difference of 5 ° C (7 - 12° C) in water and the secondary, for a temperature difference of 6 ° C (8 - 14 ° C).

For hot water, there are three secondary loops, one for the dehumidifier auxiliary water coils and the AHU units, a second one for the fan-coil units and the air curtains located in the locker rooms and a last one is responsible for heating the pool water.
The primary loop connected to the boiler is designed for a water temperature difference of 10° C (50 - 40 ° C) and the secondary, for a temperature difference of 12 ° C (50 - 38° C) in dehumidifier water coils and the AHU units loop, 9.5 ° C (50 – 40.5° C) in the

fan-coil units loop and 8 ° C (50 - 42 ° C) in the loops responsible for heating the pool water.

The heat recovery system is disposed in a primary-secondary configuration as well, with the chiller condenser side as the starting point of the primary loop and a heat exchange as an extra source of energy for the pool water in the secondary one.

For heating the pools water, the energy recovery system of the chiller is used as the first source. In case of not having enough energy or failure, a boiler would work as support unit, besides it has been mentioned that the pool water is pre-heated from the dehumidifier recovery system.

The division between the primary and secondary loops is done by implanting a separator manifold dimensioned to achieve a very low water flow rate in order to avoid interference between the circulation pumps of the two loops.

If is not established a right balancing of the system, this will consequence in unequal distribution of the flow and there will be a surplus effect in some of the components so that will not be ensured a correct heating/chilling in all parts of the installation. For that reason the distribution of the water lines has direct return, provided with the necessary balancing valves in each of the line to ensure the correct distribution of the flow in the system. In addition, each line has its own stopcock and partial drain tap to allow the line isolation because of maintenance needs or operating conditions of the building.

In a first stage the energy loss in the pool enclosure and the dehumidification needs will be calculated to select a dehumidifier system.

In a second stage, the energy loss through the building will be calculated with computer software, to know the power required by the chiller and the boiler. Although the HVAC system will not be study, it is indispensable this calculation to know the recovery energy that can be obtained from the chiller.

In the third stage the system equipment will be selected, dehumidifiers, chiller, heat exchangers, pumps and other hydraulic components.

In the last phase the selection of the components will be justified from an energy efficiency view making a comparison with a solar system.

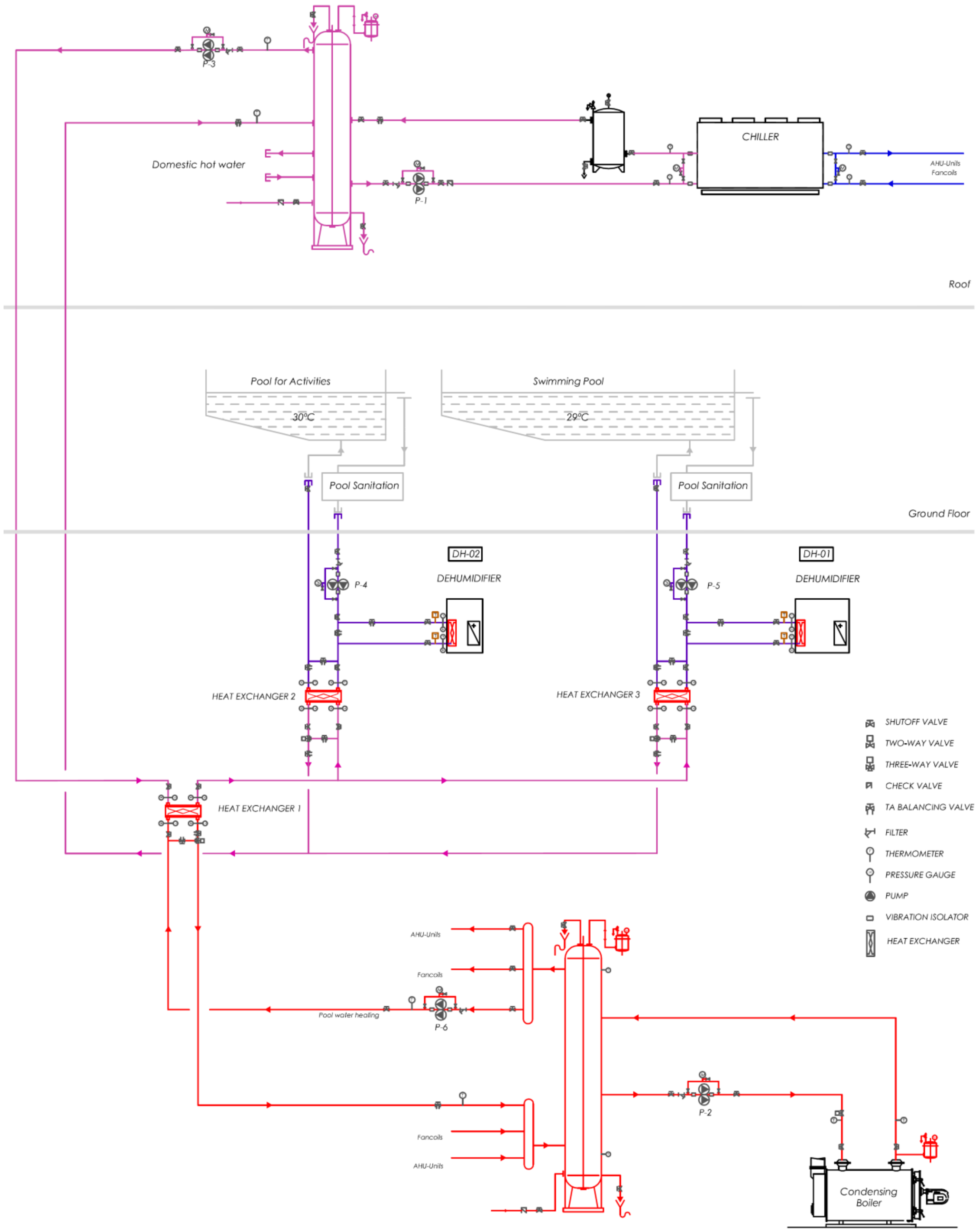

Figure 3.4. System diagram

3.3 Building energy demand

3.3.1 Heat loss of the pool

3.3.1.1 Calculation Assumptions and Data

Measurements: Considering the Sport centre drafts (Figure 3.5, Figure 3.6, Figure 3.7) and measuring in the Autocad document, the Tables 3.1 and 3.2 resume the pool and the enclosure dimensions:

Table 3.1. Pools dimensions

Pool	Water surface Area	Width	Depth	Volume
Swimming pool	312,5 m^2	12,5 m	1,2 m	375 m^3
Pool for activities	75 m^2	6 m	1 m	75 m^3

Table 3.2. Pool enclosures dimensions

Pool enclosure	Wall	Windows	Roof	LNC Floor	Dividing Wall
Swimming pool	185 m^2	79 m^2	609 m^2	541 m^2	148 m^2
Pool for activities	130 m^2	11 m^2	230 m^2	232 m^2	105 m^2

The total volume of the pool enclosure is 3130 m^3.

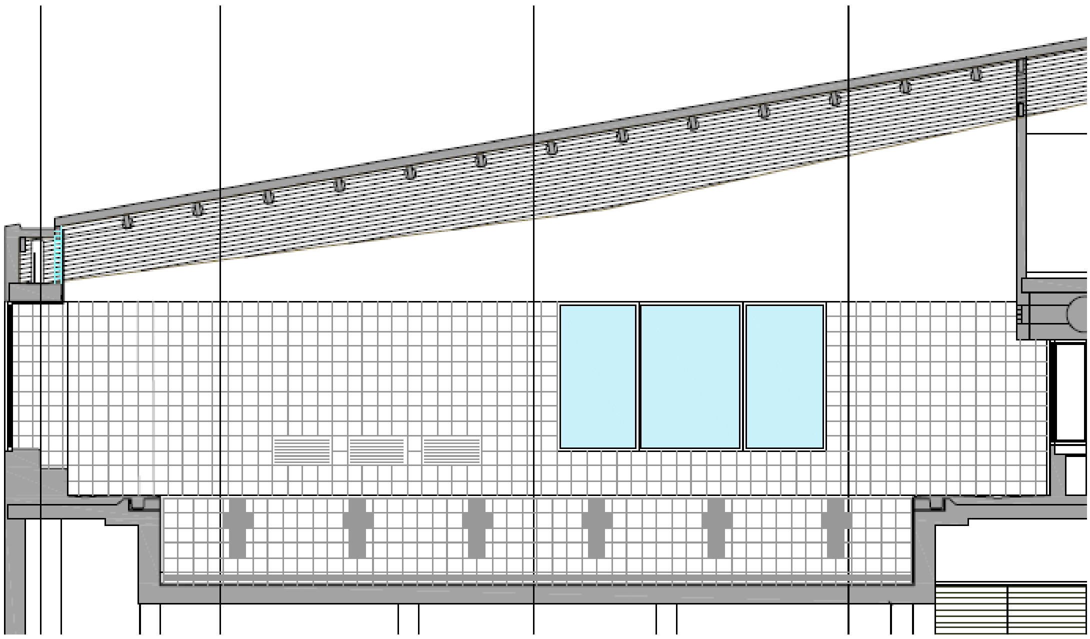

Figure 3.5. Section of the pool enclosure

Figure 3.6. Pool enclosure outside view

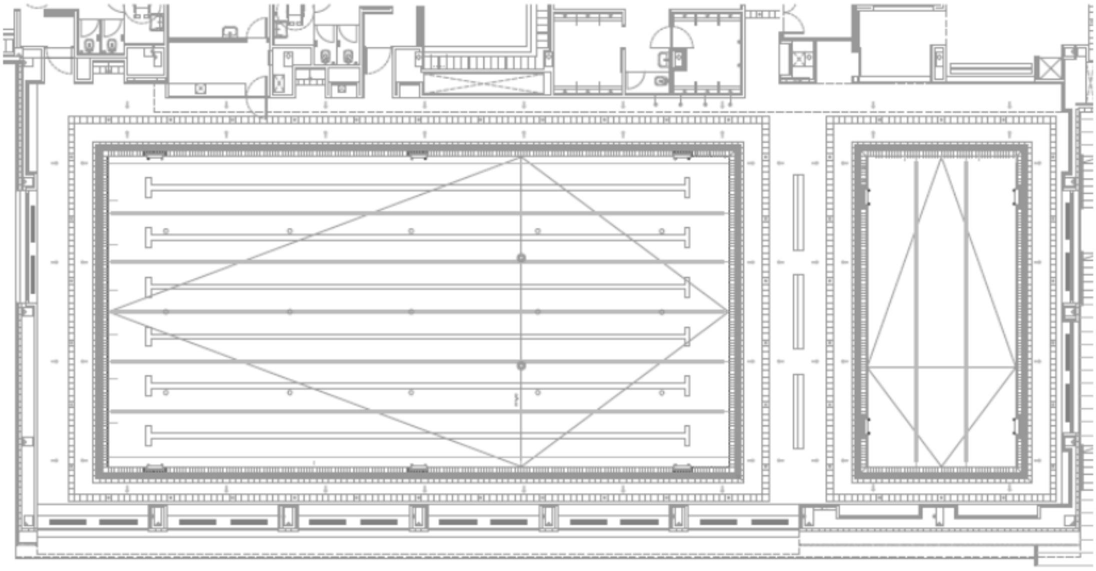

Figure 3.7. Pool floor view

Materials and Heat Thermal Transmittance: In the absence of material data in the architectural project we will use the minimum values that the Spanish Building Code allows for each enclosure part (Table 3.3).

Table 3.3. Thermal transmittance for different materials

Wall	0,66 W/ m^2 K	0,57 kcal/h m^2 K
Windows	3,00 W/ m^2 K	2,58 kcal/h m^2 K
Roof	0,38 W/ m^2 K	0,33 kcal/h m^2 K
Floor	0,49 W/ m^2 K	0,55 kcal/h m^2 K
Dividing Wall	0,86 W/ m^2 K	0,74 kcal/h m^2 K

Temperature: According to the IT.1 RITE standard [4], the air temperature will remain 1 to 3°C above the temperature of the water temperature in the pool. Likewise, the relative humidity of the environment is always maintained below 65 % to preserve the formation of condensation in enclosures. Although the maximum temperature of the air enclosure by regulation is 30ºC, a temperature of 32ºC will be used in order to assure the temperature needed after the supply air mix with the enclosure air.

Under these conditions of temperature and humidity of users comfort is assure. These temperature and humidity conditions prevent swimmers feeling cold and limit the evaporation of the water surface in the pool.

Tabla 3.4. Pool Temperature and humidity conditions

Pool	Water Temperature	Air Temperature	Relative Humidity	Pool Enclosure Volume
Swimming pool	29 °C	32 °C	65%	3.130 m^3
Pool for activities	30 °C	32 °C	65%	1.185 m^3

Weather Information: In order to fulfil the Spanish regulation, the UNE 100001:2001standard [13] will be used, which define the limit weather conditions for projects. This document has been prepared by the AEN/CTN 100 technical committee.

As the system will be projected in Segovia, Salamanca the closest city with regulated data, will be considered. The city is situated 1002 m above the sea level with latitude of 41 º.The following extreme weather conditions will be used (Table 3.5).

Table 3.5. Outside extreme weather conditions

Summer	
Outside Temperature	32,4 °C
Outside Relative Humidity	36,5%
Winter	
Water temperature supply	10ºC
Outside Temperature	-6,3 ºC
Outside Relative Humidity	90%

Although extreme weather conditions are used in order to assure a good performance through the whole year, in the table 3.6 the monthly temperature values are exposed.

Table 3.6. Monthly weather data

Month	Temperature Average ºC	Relative Humidity Average %	Water Supply Temperature Average ºC	Number of sunshine hours	Incident Energy kWh/ m^2.month
January	4	76	4	248	49
February	6	72	5	252	68
March	10	65	7	279	115
April	12	66	9	285	153
May	15	63	10	294	175
June	20	57	11	285	188
July	24	49	12	294	221
August	23	57	11	294	214
September	20	50	10	270	156
October	14	68	9	279	98
November	9	75	7	240	56
December	5	78	4	232	43

Pool occupancy rate and Start-up time assumptions: Based in a manufacturer specification for sport centre pools, 72 hours start-up time will be assumed. The occupancy rate is shown below.

- 4 h with 0.2 swimmers/m^2.h
- 5 h with 0.15 swimmers/m^2.h
- 6 h with 0.1 swimmers/m^2.h
- 9 h with 0 swimmers/m^2.h

3.3.1.2 Heat loss of the pools

The heat loss of the pools and their enclosure can be summarized in the Table 3.7 and Table 3.8:

Table 3.7. Heat loss of the pools

Water Heat Loss	Swimming Pool	Pool for activities
Evaporation	40736 W	12351 W
Radiation	2640 W	650 W
Convection	-845 W	-118 W
Water heating	8630 W	1817 W
Conduction	8871 W	2598 W

Table 3.8. Heat loss of the enclosure

Enclosure Heat Loss	Swimming Pool	Pool for activities
Conduction	27620 W	10620 W
Air heating	33230 W	7980 W
Total	60850 W	18600 W

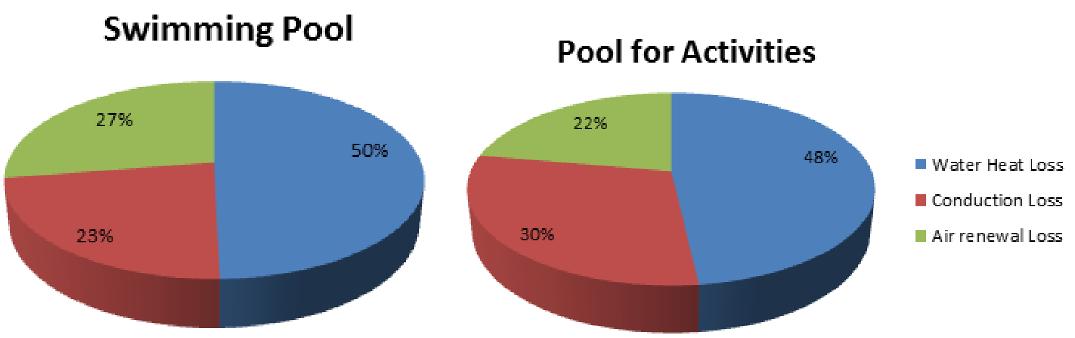

Figure 3.8. Share of heating losses

3.3.2 Building Load Calculations

Every engineer needs to have handy empirical data for calculating approximates loads and equipment sizes through the early stages of the design process. However, in a more developed phase is essential to make detailed and well-documented load calculations, because are the key for a right equipment selection, duct or pipes design, and psychrometric analysis.

3.3.2.1 Design Criteria

Indoor design conditions are determined by comfort or process requirements. According to the RITE (IT 1.1.4.1.2) [11], the following design temperature limits for occupied zones are considered in calculating the project loads.

- Winter: 22 ° C, with an operating temperature between 21 ° C and 23 ° C and a relative humidity between 40%-50%. In extreme weather conditions and start-up periods should not be considered relative humidity values of less than 35%.

- Summer: 24 ° C, with an operating temperature between 23 ° and 25 ° C and a relative humidity between 45-60 %.

These two values of temperature must be maintained within the occupied zone. The occupied zone is considered as the volume intended for human occupancy and is defined by vertical planes parallel to the walls of the room and a horizontal plane that delimits the height of the floor. The distances of these planes from the inner surfaces of the room are as follows:

- Lower limit from ground: 5 cm.
- Upper limit from the ground: 180 cm.
- Outer walls with windows or doors: 100 cm.
- Inner walls and outer walls without windows: 50 cm.
- Doors and transit zones: 100 cm.

It will not be considered as occupied zones, those that may have significant temperature variations from the average or may have significant airflow rates around people , as transit areas or areas near doors frequently used, zones close to any unit which impulse air or with high heat production. Outside design conditions are determined from regulated data for the specific location, based on weather records.

Once the criteria is defined and the HAP from Carrier method is chosen the load calculations are a matter of accurately determining areas, internal load densities such as people, process loads, lighting and any unusual conditions.

Depending on the zone orientation peak loads will vary with the hour and season. In order to size the air-handling equipment, HAP calculates the loads in accordance with the peak annual load in that zone thanks to the database incorporated. Some specific requirements will be apply in the rooms with more activity following RITE specificacions [4].

- Activity Rooms
  - People load:
    - $Q_{sensible}$ = 154 W/person
    - Q_{latent} = 271 W/person
    - Ocuppancy rate: 3 m^2/person
  - Outdoor air flow rate per person:
    - Qextair= 28,8 m^3/h/person
  - Lighthing and eletrical devices:
    - Lighthing (10W/ m^2).
    - 3000W Audio equipment.

- Spinning Room
  - People load:
    - $Q_{sensible}$ = 308 W/person
    - Q_{latent} = 435 W/person
    - Ocuppancy rate: Number of bicycles ⬄ 2 m^2/person
  - Outdoor air flow rate per person:
    - Qextair = 28,8 m^3/h/person
  - Lighthing and eletrical devices:
    - Lighthing (10W/ m^2).
    - 3000W Audio equipment.

- Fitness Room
  - People load:
    - $Q_{sensible}$ = 154 W/per
    - Q_{latent} = 271 W/per
  - Ocuppancy rate: Workout equipment ⬄ 5,3 m^2/per
  - Outdoor air flow rate per person:
    - Qaext= 28,8 m3/h/per
  - Lighthing and eletrical devices:
    - Lighthing (10W/ m^2).
    - 1200W Audio equipment.

In order to dothe load calculations ,it will be used the space distribution specified in Appendix C and considering the occupied zones which are expected to be climatized.

In the summary table 3.30 it is shown the output data provided by the HAP Carrier's software for all the spaces. With the input data the software calculates the window and solar loads, the wall, windows, doors and roof/floor transmission loss or internal loads as lighting, electric equipment and people. The return fan, ventilation and supply fan loads are estimated in accordance with the ventilation requirements.

A total cooling load of 217 kW is calculated, it will be considered 225 kW in order to select the chiller, assuming an additional 5% because of the AHU-units and fancoils power installed.

The total heating load is 161 kW, it will be considered 170 kW in order to select the boiler because of the AHU-units and fancoils power installed should be higher.

Table 3.9. Heating and cooling loads of spaces

	Sensible Cooling	Latent Cooling	Sensible Heating	Latent Heating
Zone 1	15808 W	11913 W	5979 W	5587 W
Zone 2	33245 W	17610 W	22259 W	20039 W
Zone 3	12868 W	7185 W	4787 W	2760 W
Zone 4	49811 W	22502 W	11979 W	8102 W
Zone 5	15815 W	8464 W	7456 W	2997 W
Zone 6	1801 W	306 W	1050 W	0 W
Zone 7	1981 W	284 W	2088 W	0 W
Zone 8	239 W	120 W	270 W	0 W
Zone 9	239 W	120 W	270 W	0 W
Zone 10	461 W	67 W	289 W	0 W
Zone 11	876 W	119 W	283 W	0 W
Zone 12	2945 W	274 W	1829 W	0 W
Zone 12+	1483 W	209 W	1496 W	0 W
Zone 13	1335 W	179 W	266 W	0 W
Zone 14	3053 W	0 W	1278 W	0 W
Zone 15	322 W	97 W	715 W	0 W
Zone 16	2283 W	502 W	2675 W	0 W
Zone 17	1655 W	337 W	908 W	0 W
Zone 18	-	-	22142 W	0 W
Zone 19	-	-	20733 W	0 W
Zone 20	-	-	12754 W	0 W
TOTAL	146220 W	70288 W	121506 W	39485 W

3.4 Swimming Pool Dehumidification System

In accordance with the Bernier equation [5] the evaporated mass flow per hour defines the dehumidification capacity needed by the system (Table 3.31).

Table 3.9. Dehumidification needs

	Swimming Pool	Pool for activities
Dehumidification needs	86,45 kg/h	25,50 kg/h

In order to fulfill our dehumidification needs, a software developed by Sedical, a leading company in energy saving, is used to do a more accurate selection, taking into account other factors given by the experience of this company in the field to guarantee top levels of comfort all year round.

Taking into consideration the specific requirements of the project, a system that dehumidify, cool, heat and renew the ambient air whilst simultaneously heating the pool water free of charge is chosen.

The Dry-Pool dehumidifier is an air and/or water cooled machine with hermetic scroll compressors and R410A refrigerant gas that has been designed to meet all the necessities of the swimming pool environment. Thanks to the use of the refrigerant circuit, the air treatment energy loads are drastically reduced. An auxiliary hot water coil is installed after the condensing coil, that is supplied with a motor-driven three-way valve and balancing valve on the bypass branch completely managed by a microprocessor.

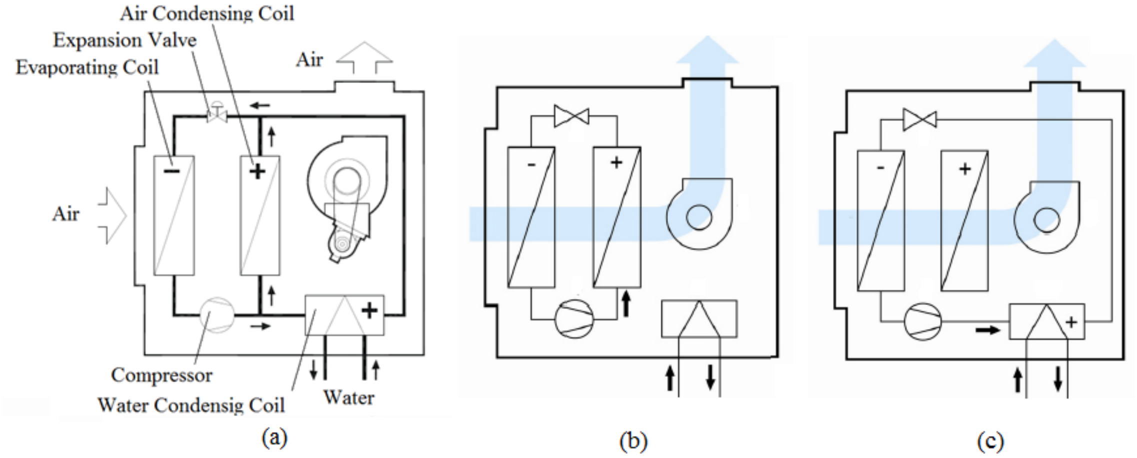

Figure 3.9. Dehumidifier diagrams

The Figure 3.9 shows a schematic dehumidifier with some of its main components: compressor, evaporating and condensing coils and the expansion valve. It is possible to heat the swimming pool water by recovering all the energy generated during the dehumidification phase, leading to a noticeable reduction in the pool water heating costs.
In the diagram (b) the water coil is not operating and the refrigerant only flows through the coil intended to heat the air, providing 100% of the recovery heat directly to the pool air. In the diagram (c) it is shown the opposite case, where the water coil is responsible for heating the water from the pool.

The evaporation coil has copper pipes and copper fins, guarantying reliability even in aggressive environments such as the swimming pool. In the diagram (a) there are two condensing coils designed in parallel, about 45% of the recovery heat is removed in the water coil, and 55% in the air coil.

The operation is simple, the air from the pool flows through the evaporator where it is cooled and dehumidified by condensing water. The condensation heat generated is used to heat the same air in the condensing coil and the water from the pool. The dehumidifier is capable to distribute the refrigerant to the coil responsible for heating the air or the one responsible for heating the water, but it is possible to heat only the air if the extreme conditions require it.

The difference with a conventional air/air chiller is that the chiller would be able to cool the air and dehumidify it but losing 100% of the condensation heat dissipating to the environment. A dehumidifier uses 100% of the condensation heat. This recovery heat from condensing coils is a source of residual energy that the Spanish Technical Building Code [12] considers in order to reduce the minimum solar contribution.

DAESY-DRESY-DTESY-DEESY		128	271	281	294
Compressors/Circuits	No.	1	2	2	2
❶ DRESY dehumidification capacity	l/h	27,4	71,0	80,7	93,7
❶ Net cooling capacity transferred to the air 55%	kW	13,1	35,7	46,6	50,8
❶ Heating capacity transferred to the water 45%	kW	19,6	56,1	61,0	69,7
❷ Heating capacity transferred to the air 100%	kW	43,7	116,4	121,7	143,2
❶ Total absorbed power	kW	8,3	21,8	25,2	28,5
❸ DTESY dehumidification capacity	l/h	27,4	70,5	80,0	92,0
❸ Net cooling capacity transferred to the air	kW	36,9	96,9	101,7	119,8
❸ DTESY heating capacity transferred to the water 100%	kW	45,4	119,3	127,6	149,2
Nominal air flow	m³/h	6.200	16.500	18.000	21.000
Width	mm	850	850	850	1.230
Depth	mm	1.600	2.270	2.270	3.370
Height	mm	1.890	1.890	1.890	2.000

Figure 3.10. Different dehumidifiers models

The air conditioning of a closed pool has very different requirements from those of a typical system for residential or commercial utilities. When selecting reasonable operating conditions for swimming pool environments, it is important to find the right balance between the comfort of the swimming pool users and the energy costs involved in running the system.

In order to minimize the water evaporation rate and thus the system running costs, it is always necessary to have an air temperature that is higher than the temperature of the water in the pool.

The greater the temperatures difference between air and water, the lower the evaporation rate. Obviously, for reasons of comfort, this difference cannot be excessive and the manufacturer usually advises keeping it at a Δt of 2 /3 °C.

The values typically used for this type of system are listed below:

- Ambient air temperature: 27÷32°C
- Ambient air relative humidity: 60÷70%
- Pool water temperature: 26 to 30°C

In figure 3.10 it is shown the different dehumidifier models (model 128, model 271, etc) with different specifications. Following the guidelines from Sedical, the model DRESY 281 [15] is selected for the swimming pool and DRESY 128 [15] for the pool of activities.

Although the dehumidification capacity is a little smaller than the dehumidification needs, Sedical recommend that selection because it will be the maximum rate only in a small period of time in the warmest months.

Table 3.10. Energy needs and dehumidifier specification comparison

Enclosure and Water Energy Summary	
Dehumidification needs	86,45 kg/h
Start-up Power	238,48 kW
Water Average Loss	60,03 kW
Water Peak Loss	67,83 kW
Enclosure Loss	60,85 kW
Dehumidification System: DRESY 281	
Dehumidification Capacity	80,7 kg/h
Air Flow rate (32 ºC; 65%; 0,89 m^3/ Kg dry air)	18000m^3/h
Heating Capacity transferred to the water	61,0 kW
Heating Capacity transferred to the air	67,7 kW
Refrigeration Capacity in the Evaporation Coil	101,5 kW

According the heat loss in the water and the enclosure and the dehumidification needs calculated before, the software shows the water average loss during the day, pondering the daytime hours at full occupancy, and without occupancy at night. Considering the swimming pool dehumidifier, it will be done an approach to its operation in the psychrometric chart.

The dehumidifier would be able to control the humidity of the air in the enclosure and keep the temperature of the enclosure and the water in optimum levels (Table 3.10). It would be needed a boiler for heating the pool water and the enclosure air in the system start-up.

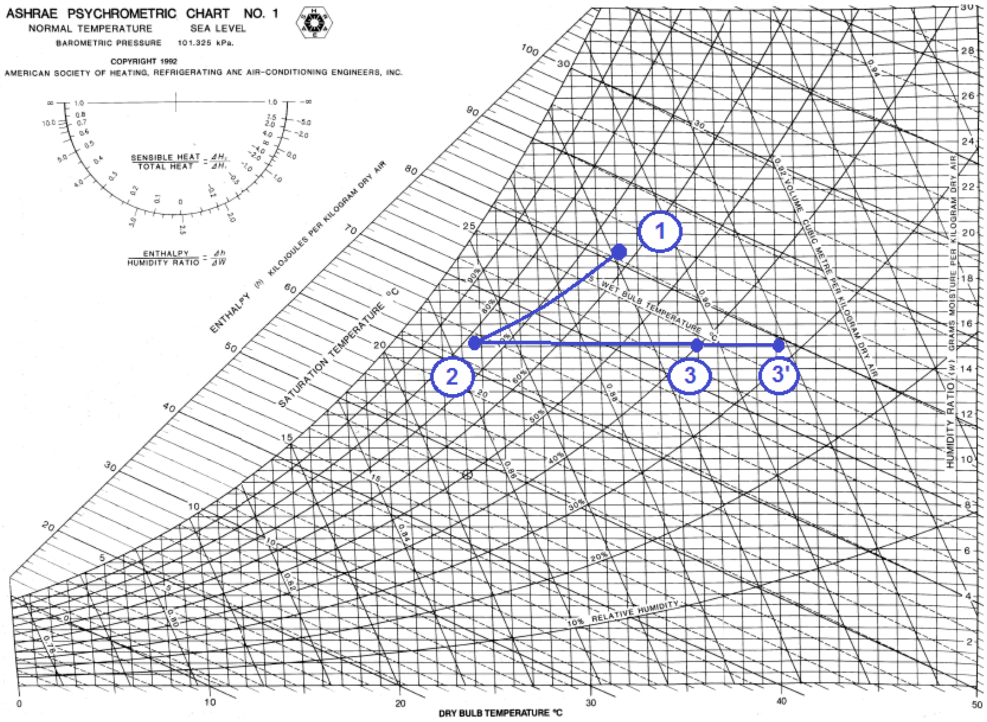

Figure 3.11. Dehumidifier performance in the psychrometric chart

For this case, using our dehumidifier as a conventional chiller air to air able to remove 80,7 kg /h of moisture in an 18000 m^3/h air flow rate:

At the point 1 in the Figure 3.11, the return air (32 ºC; 65%; 0,89 m^3/ kg dry air), the dry air mass flow would be 20224 kg/h dry air.

Dividing the dehumidification capacity by the dry air mass, it is calculated the water mass removed from the air (4 g of water/ kg of dry air)

With a refrigeration capacity in the evaporation coil of 101,5 kW, the enthalpy change between the inlet and outlet air flow rate is calculated with the equation:

$$P = \dot{m}\,(h_2 - h_1) \qquad (20)$$

Table 3.11. Thermodynamic properties of points 1 and 2

Point	Data		Psychrometric Chart	
Point 1	32 ºC	65%	$19{,}37\frac{\text{g of water}}{\text{kg dry air}}$	$64{,}32\frac{\text{kJ}}{\text{kg dry air}}$
Point 2	$15{,}37\frac{\text{g of water}}{\text{kg dry air}}$	$62{,}92\frac{\text{kJ}}{\text{kg dry air}}$	24 ºC	78%

The dehumidifier works as a chiller air /air with the outlet flow rate at point 2 and 101,5 kW refrigeration capacity.

In the psychrometric chart is known that the horizontal component of the condition line is the change in sensible heat while the vertical component represents the change in latent heat.

The sensible heat in the cooling 1-2 is 44,94 kW and the remaining heat is used to dehumidify from $19{,}37\frac{\text{g of water}}{\text{kg dry air}}$ to $15{,}37\frac{\text{g of water}}{\text{kg dry air}}$

The Dresy dehumidifier [15] also transfers 67,7 kW to the pool air turning the point 2 into the point 3, and transfers 61 kW to the pool water. Moreover, it can operates, if necessary, transferring all the heat to the air, in which case the assigned heat would be 128,7 kW and the point 3 would become 3'.

Clearly the dehumidifier takes advantage of this residual energy source and is designed to recover that heat (Figure 3.12). It does not make sense to heat the water of the pool with solar energy and dissipate the heat produced in the dehumidification process.

In the calculations above, it was considered that the dehumidifier inlet air is completely returned from the pool enclosure at 32 ° C and 65%, but the reality is that the intake air is a mixture of indoor air and outdoor air with an approximate ratio of 80/20%, as it is showed in the picture, and this causes the inlet air is 1 to 3 ° C lower than the air from the enclosure. The dehumidifier sends it back to the pool dehumidified and at a temperature about 3 ° C higher than the inlet air. In order to balance out the conduction losses in windows and walls it was selected 32ºC as reference to maintain the temperature in the pool not lower than 30 ºC.

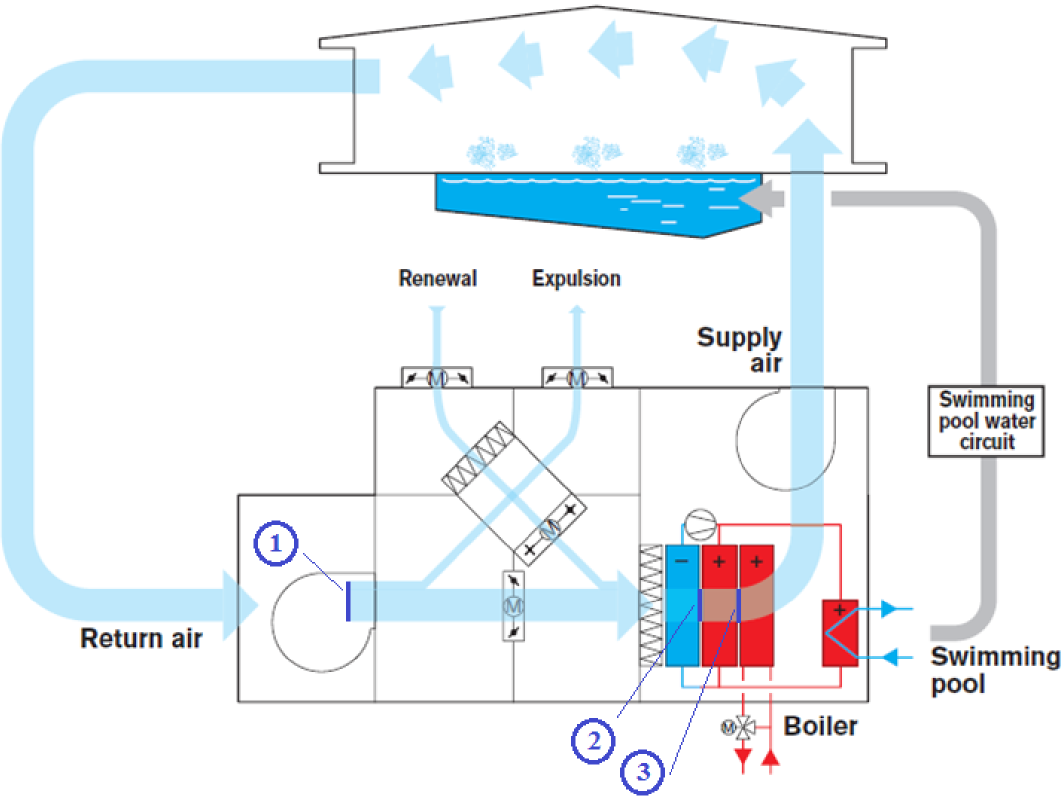

Figure 3.12. Dehumidifier performance

3.5 Air diffusion in the pool enclosure

Distribute the air in the pool enclosure is very complex due to several factors:
The pool enclosure is more than 4 meters high with a variable height, so a system that discharges air jets at high speed horizontally to create a barrier to prevent air stratification is needed. This is crucial to maintain the warm air close to the occupied zone and avoid an accumulation in higher zones.

The pool enclosure is quite big and the air can only be dispersed through the walls because, for architectural reasons, the ceiling cannot hold a duct structure. It is necessary to disperse the supply air far enough to cover the pool surface.

Since exterior windows are the main condensation concern, the supply air must be focused there, as it is shown in Figure 3.13. The warm air from the dehumidifier will keep the window surface temperature above the dew point to guarantee no condensation problems.

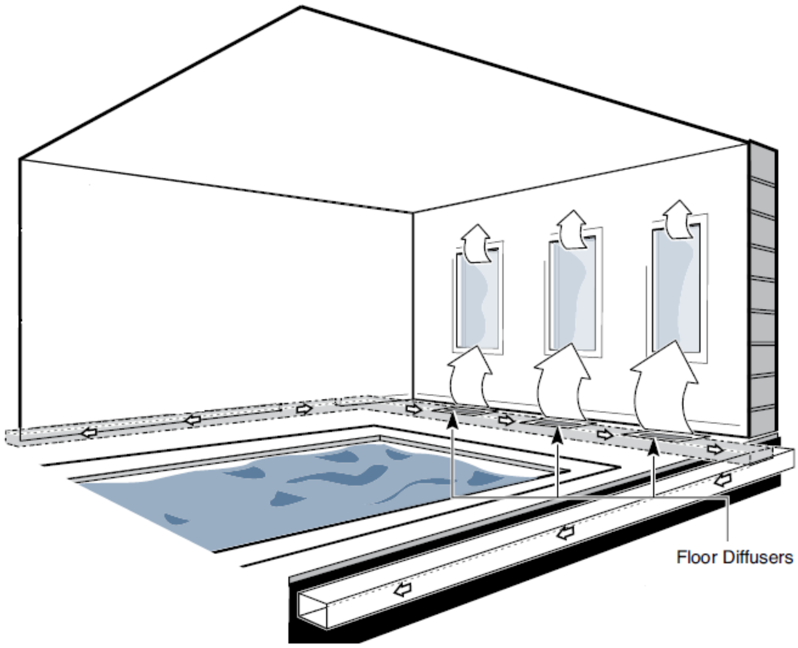

Figure 3.13. Diffusers for pool windows

The ceiling has an architectural structure in which condensation can be produced with the consequently serious damage in the materials and coatings, so is important to impulse air there as well.

It is decided to perform a combined diffusion of air, consisting of nozzles for treatment of the general air avoiding condensation on ceilings and linear floor diffusers under the windows.

To ensure the air barrier above the pool the DF-49 nozzles from Koolair [16] are used, since they provide long throws with a low noise level, releasing a long air jet to a length of over 30 meters. These long-throw, high-flow diffusers are particularly useful when the air jet should reach some distance specially recommended for sport centers.

The airflow needed to cover the pools area is 5,700 m^3 / h through 10 nozzles (570 m^3 / h per nozzle), installed at a height of 3,4 m. The selection is made for a temperature difference between ambient and renewal air of 10 °C.

The diffuser should be selected to obtain the following values:

- Maximum velocity in the occupied area: 0,2 m/s mandatory by IT 1.1.4.1.3 (RITE) [4]
- The vertical temperature gradient must not exceed 3°C [4].
- The sound power level of the selected equipment must not exceed 40 dB(A).

The selection is done using the catalogue charts provided by the manufacturer [16] and the initial data (Table 3.12).

Table 3.12. Initial diffusion data

Reach distance	11 m
Height from floor	4 m
Nozzle flow 570 m^3 / h	570 m^3 / h
Supply Temperature	39° C
Room Temperature	29° C
Difference of temperatures ΔT	10° C
Height of occupied area	2 m

The sound power limit of 40 dB(A) must not be exceeded, so the size 10 is preselected with the quick selection table and that maximum value.

Using this size for 570 m^3 / h, the pressure drop and real sound power level is determined with the chart (Figure 1 Appendix D).

Using the velocity of the air jet for the throw chart (Figure 2 Appendix D) with an supply angle of -4°, the throw and velocity are determined.

The impact point under isothermal conditions is calculated with a simple trigonometric relation. With the temperature difference, the vertical deviation caused by a temperature difference between the supply and ambient air is view in the chart (Figure 3.16) and the impact point height from the floor of the air jets is calculated as the difference between the height at isothermal conditions and the deviation.

With the ratio between air flow velocities chart (Figure 3 Appendix D) the air velocity in the occupied area is determined.

The temperature of the air jet at its inlet in the occupied zone is determined by the chart (Figure 4 Appendix D) and the induction rate, quotient between the air flow for a throw X and the air flow supplied in the zone, determined by the chart (Figure 5 Appendix D). The results of the diffusion study are shown in the table 3.13 below. According to the data obtained, it is ensured that the entire surface pool is covered, without any risk of condensation in the ceiling structure and average speeds in occupied area are within the limits of the comfort conditions. The air jet is represented in the Figure 3.14.

Table 3.13. Diffusion results summary

Effective supply velocity	12,5 m/s
Size	10
Total pressure drop	89 Pa
Sound power level	38 dB
Velocity of the jets at throw X	0,78 m/s
Deviation of the air jet	1,3
Velocity of the jets in the occupied area	0,08 m/s
Induction rate	30,9
Temperature in occupied area	29,5 ºC

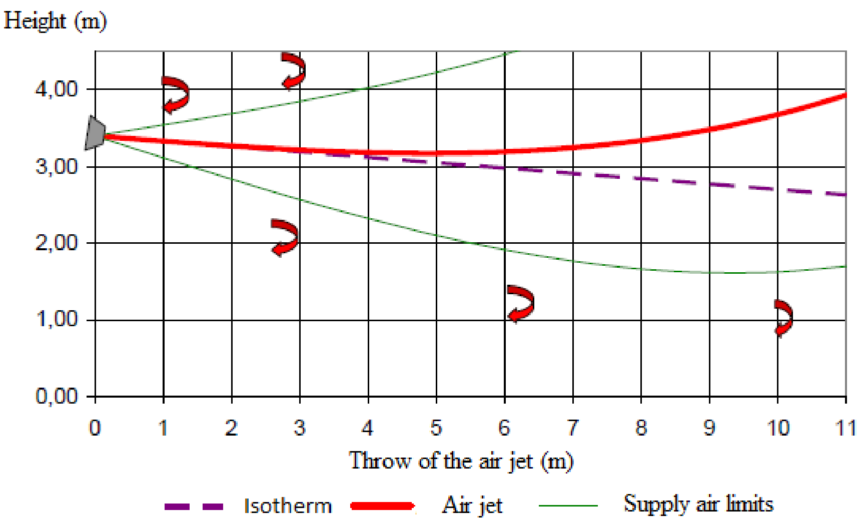

Figure 3.14. Throw air graphic

3.6 Airduct Design

The purpose of a duct system is to carry air from the AHU units or in this case a dehumidifier to the zone to be conditioned. The duct is the static component of the installation through which air flows within the building.

To fulfill this task in a practical way, the system is designed satisfying some requirements as fitting in the available space, low friction loss, velocity limitations to avoid high noise levels, heat losses and gains, fire and smoke control and balancing [20].

It must be considered these practical design criteria to avoid an incorrect operation, a system with lack of sound attenuation or high air velocity can produce excessive noise levels, high friction loss leads to an inadequate air flow rate at the units producing discomfort or inefficiencies in the equipment.

The space available in ceiling plenums or obstructions such as walls or beams for the supply and return air ducts, and their appearance usually determines the system design regardless if it is not the best size from a cost or friction loss perspective. In addition, there are other building system components that interact with the duct network, so coordination is necessary in order to avoid problems in the assembly.

In the swimming pool ceiling there is not plenum and the ductwork has to be exposed and attached some way to the ceiling. For this case, rectangular ductwork is ideal because is constructed to give the appearance of a beam, furthermore rectangular ductwork is probably the most economical design for this application.

In any duct section thru which air is flowing, there is a continuous loss of pressure. This loss is called duct friction loss and depends on the following:

- Air velocity
- Duct size
- Interior surface
- Duct length
- Changes in cross sections, corners, vents, etc.

Varying any one of these factors above influences it in the ductwork. The relation of these factors is illustrated in the following equation [20] :

$$\Delta P = 0.03f\left(\frac{L}{d^{1.22}}\right)\left(\frac{V}{1000}\right)^{1.192} \qquad (21)$$

Where:

- ΔP is the friction loss
- f is the interior surface roughness (0,9 for galvanized duct)
- L is the length of the duct
- d is the duct equivalent diameter for rectangular ductwork
- V is the air velocity

This equation is used to construct the chart (Figure 3.15) based on galvanized duct.

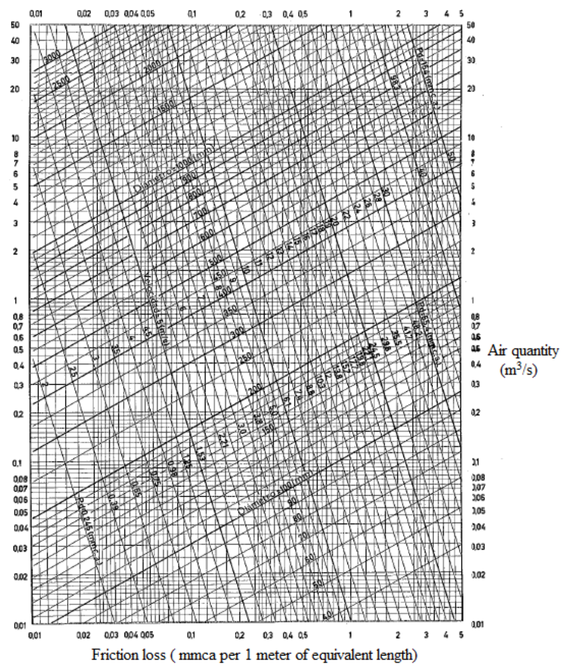

Figure 3.15. Friction loss for round duct

The usual procedure for designing a duct system is to keep a simple layout with a symmetrical position. It can be followed different methods to design a proper low velocity air system; velocity reduction, equal friction or static regain [4].

The equal friction method is the most extended and recommended, this method employs the same friction loss per foot of length for the entire system. The usual procedure is to choose an initial value for the air velocity in the main duct, smaller than the maximum recommended in the table 3.14.

Table 3.14. Recommended maximum duct velocities (m/s) [4]

APPLICATION	CONTROLLING FACTOR NOISE GENERATION Main Ducts	CONTROLLING FACTOR—DUCT FRICTION			
		Main Ducts		Branch Ducts	
		Supply	Return	Supply	Return
Residences	3	5	4	3	3
Apartments Hotel Bedrooms Hospital Bedrooms	6	7,5	6,5	6	5
Private Offices Directors Rooms Libraries	6	10	7,5	8	6
Theatres Auditoriums	4	6,5	5,5	6	4
General Offices High Class Restaurants High Class Stores Banks	7,5	10	7,5	8	6
Average Store Cafeterias	9	10	7,5	8	6
Industrial	12,5	15	9	11	7,5

The equal friction method does not satisfy the design criteria of uniform pressure at all branches and air terminals. For this reason, it is necessary to install a splitter dampers to regulate the flow to the branch or diffusor with a properly control device to regulate at each terminal for air distribution.

In order to design the ductwork, the following variables and values will be taken into consideration:

- The maximum supply air quantity required for removed the moisture in the pool and climatize the enclosure.

- The design velocity plays an important role in a low velocity system in order to avoid noises and usually is based on experience. A very high velocity results in smaller ducts and lower duct material cost but it requires a higher operating cost and bigger fan motors. A velocity of 9 m/s is selected with sound level as limiting factor in the duct section from the fan to the first branch. Then the friction loss is calculated and kept as reference throughout the system.

- The friction rate will not exceed 1,08Pa/m (0,11 mmca/m), but as the ducts are limited by the shafts dimensions and space available the friction loss will be maintained within limits. Unlike the equal friction method, with this method a maximum value is defined but if the velocity is to high the duct size is increased to avoid excessive noise in the ducts close to the diffusers or the pool enclosure. A splinter damper is installed previously to the diffusers to regulate the flow and equilibrate the system.

The calculation of the duct dimensions are represented in the tables 3.15 and 3.16 where it can be appreciated that the friction loss is maintained within the limit and the velocity of the air close to the diffusers or pool enclosure is conserved below 5m/s.

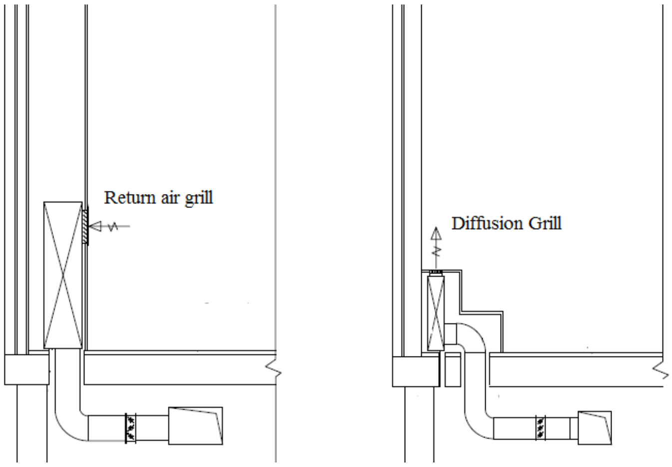

Figure 3.16. Diffusion and return grills disposal

The air return duct pulls air back through plenums located in the wall extracting the air from the pool to the dehumidifiers. The diffusers under the windows are disposed as shown in the Figure 3.22.

Table 3.15. Dimensions of supply and return ducts for the swimming pool

DUCT		AIR		SECTION				FRICTION LOSS			
	Length (m)	Flow (m^3/h)	Velocity (m/s)	Dimensions B (mm)	H (mm)	Diameter eq. (mm)	Area (m^2)	Per meter (mm.c.a./m)	Elbows/Extra (mm.c.a)	Total (mm.c.a)	Aggregate (mm.c.a)
SWIMMING-POOL - SUPPLY											
1	25,0	2.580	6,1	350	350	385,56	0,117	0,11	7,00	9,84	9,84
2	3,0	570	3,3	250	200	246,02	0,048	0,06	8,90	9,09	9,09
3	4,0	1.140	4,3	400	200	307,93	0,074	0,08		0,31	9,40
4	4,0	1.710	4,8	550	200	356,49	0,100	0,08		0,32	9,72
5	3,0	570	3,3	250	200	246,02	0,048	0,06		0,19	0,19
6	4,0	1.140	4,3	400	200	307,93	0,074	0,08		0,31	0,50
7	4,0	1.710	4,8	550	200	356,49	0,100	0,08		0,32	0,82
8	5,0	3.420	4,8	600	350	501,20	0,197	0,05		0,27	9,98
9	8,0	6.000	6,5	600	450	571,23	0,256	0,08		0,63	10,61
7	10,0	2.000	4,8	350	350	385,56	0,117	0,07	7,00	7,72	7,72
8	6,0	4.000	5,8	500	400	492,04	0,190	0,08		0,46	8,18
9	6,0	6.000	6,8	650	400	558,44	0,245	0,09		0,52	8,70
10	6,0	8.000	7,5	800	400	615,86	0,298	0,09		0,55	9,25
11	10,0	2.000	4,8	350	350	385,56	0,117	0,07		0,72	0,72
12	6,0	4.000	5,8	500	400	492,04	0,190	0,08		0,46	0,46
13	25,0	12.000	8,4	1100	400	712,99	0,399	0,09		2,35	11,60
12	1,0	18.000	9,1	1200	500	837,58	0,551	0,09		0,09	11,69
SWIMMING-POOL - RETURN											
1	3,0	3.800	9,0	350	350	385,56	0,117	0,23	7,00	7,69	7,69
2	20,0	11.400	7,9	850	500	713,17	0,399	0,09		1,71	9,40
3	6,0	13.600	9,5	850	500	713,17	0,399	0,12		0,71	10,11
4	8,0	14.700	10,2	850	500	713,17	0,399	0,14		1,09	11,20
5	5,0	18.000	9,1	1200	500	837,58	0,551	0,09		0,45	11,65
SWIMMING-POOL - OUTDOOR											
	25,0	18.000	10,4	850	600	784,33	0,483	0,12	5,00	8,09	8,09
SWIMMING-POOL - EXHAUST											
	25,0	18.000	10,4	850	600	784,33	0,483	0,12	5,00	8,09	8,09

Table 3.16. Dimensions of supply and return ducts for the pool of activities

DUCT		AIR		SECTION				FRICTION LOSS			
	Length (m)	Flow (m^3/h)	Velocity (m/s)	Dimensions B (mm)	H (mm)	Diameter eq. (mm)	Area (m^2)	Per meter (mm.c.a./m)	Elbows/Extra (mm.c.a)	Total (mm.c.a)	Aggregate (mm.c.a)
POOL FOR ACTIVITIES - SUPPLY											
1	10,0	2.245	5,3	350	350	385,56	0,117	0,09	7,00	7,88	7,88
2	25,0	4.490	6,0	550	400	515,39	0,209	0,08		1,90	9,79
3	3,0	570	3,3	250	200	246,02	0,048	0,06	8,90	9,09	9,09
4	4,0	1.140	4,3	400	200	307,93	0,074	0,08		0,31	9,40
5	11,5	1.710	4,8	550	200	356,49	0,100	0,08		0,91	10,31
6	15,0	1.710	4,1	350	350	385,56	0,117	0,05	5,00	5,81	16,11
7	7,0	6.200	8,3	550	400	515,39	0,209	0,14		0,96	17,07
POOL FOR ACTIVITIES - RETURN											
1	10,0	3.100	7,4	350	350	385,56	0,117	0,16	7,00	8,59	8,59
2	32,0	6.200	6,7	600	450	571,23	0,256	0,08		2,66	11,25
POOL FOR ACTIVITIES - OUTDOOR											
1	25,0	6.200	7,6	600	400	537,47	0,227	0,11	5,00	7,79	7,79
POOL FOR ACTIVITIES - EXHAUST											
1	25,0	6.200	7,6	600	400	537,47	0,227	0,11	5,00	7,79	7,79

In the figures 3.17 and 3.18 is represented the ductwork layout through the basement where the dehumidifiers are located, and the pool enclosure. Through the blue and orange ducts the exhaust and supply air is carried to the dehumidifiers. Those ducts are communicated to the roof by a shaft.

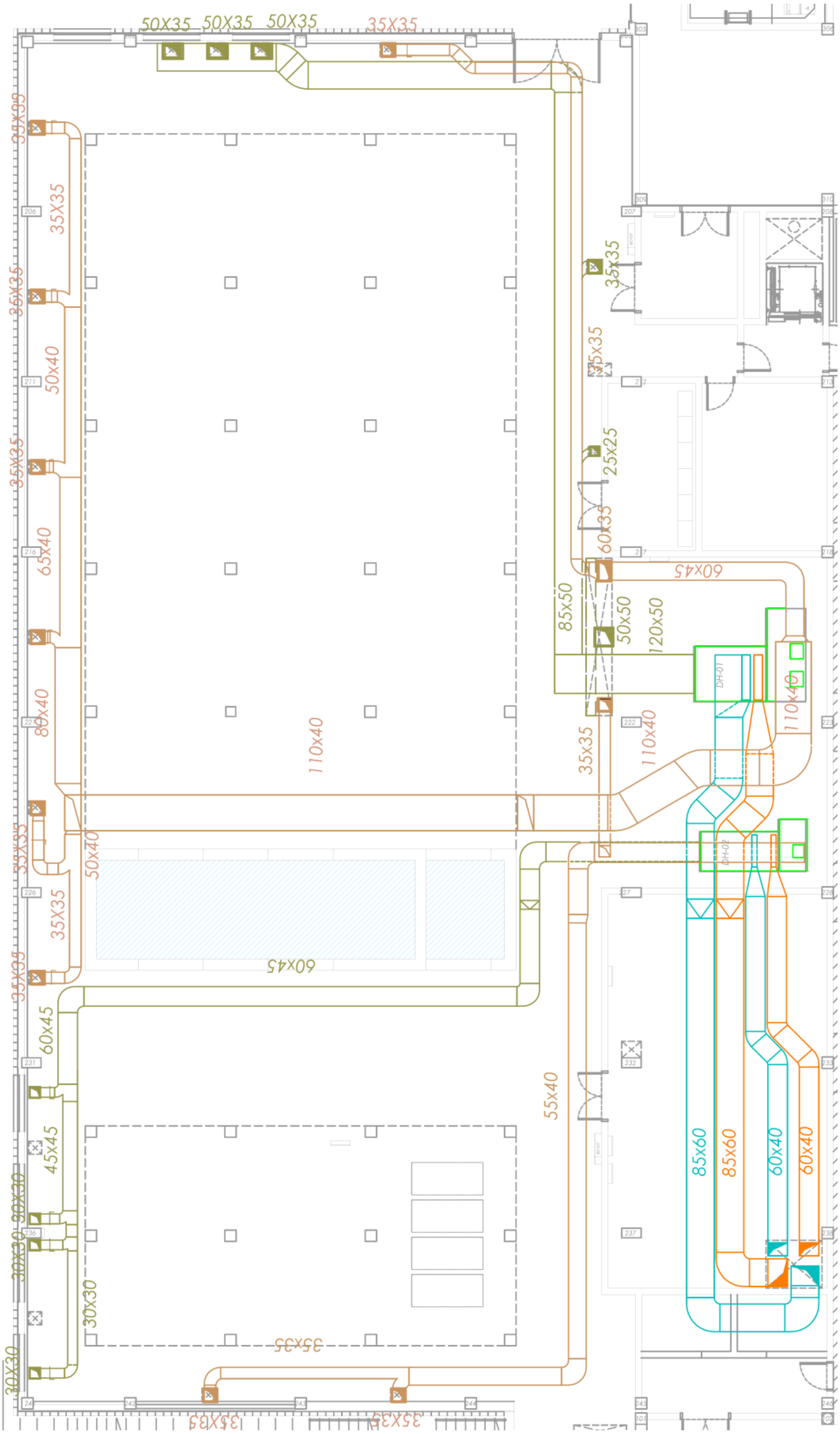

Figure 3.17. Basement air-duct distribution

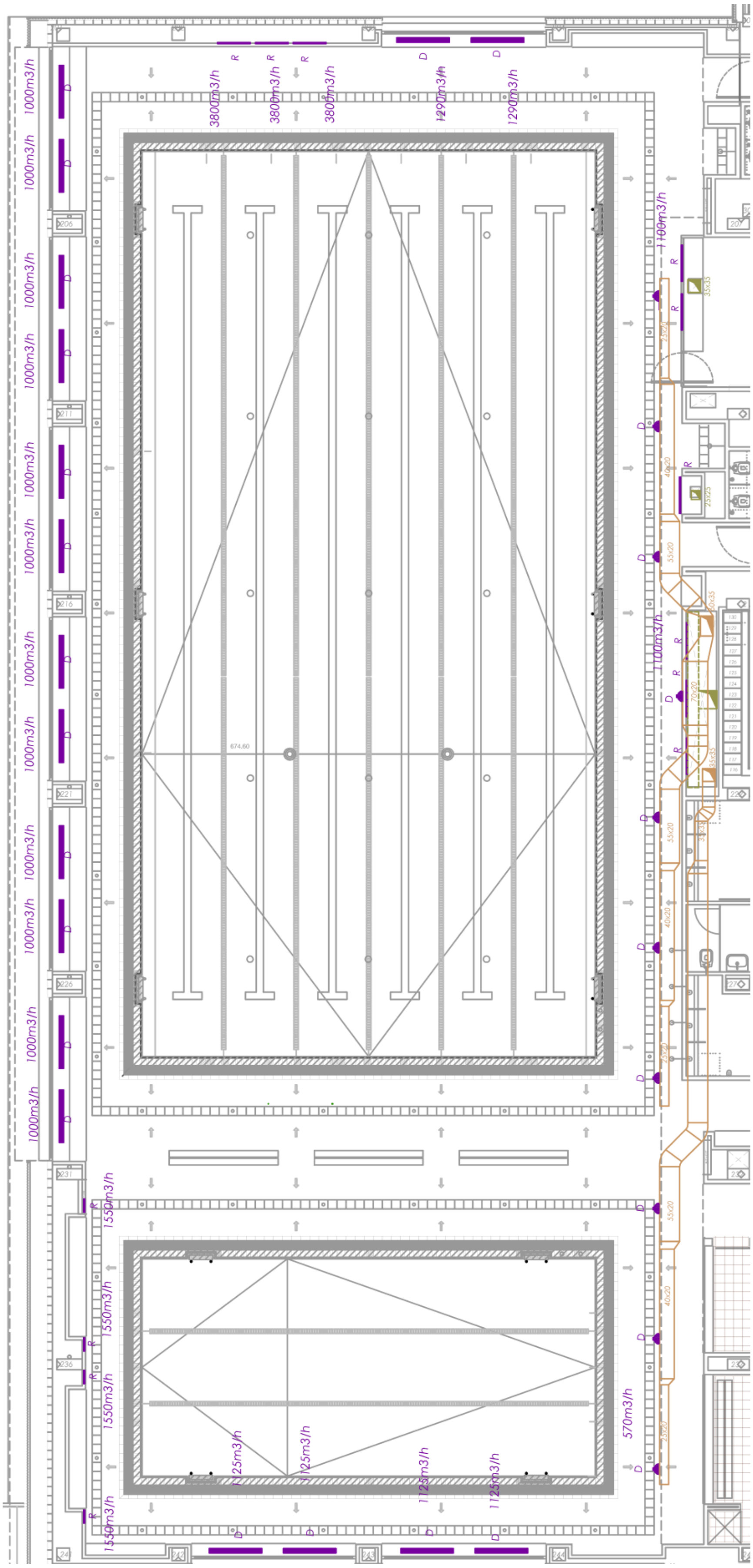

Figure 3.18. Pool enclosure duct distribution and diffusors

3.7 Chiller

The AHU-units and fancoils responsible for climatizing the sport center require a machine for the production of chilled water. The heat collected by the chiller during cooling is rejected through the condenser, however, if there is a simultaneous need for heat, then this heat can be recovered and utilized rather than being rejected outdoor.

The load calculation was intended to select a chiller with an exhaust air recovery system. Once selected, with the data provided by the manufacturer it is essential to find out the amount of energy that is capable of recover. The exhaust heat can be recovered to heat the pool water and for this reason the scope of the selection will be to fulfill the regulations and design the hydraulic recover pipelines.

Figure 3.19. Chiller

A high energy efficiency chiller with axial fans, Scroll compressors and plate type exchangers installed outdoors and air-cooled will be used (Figure 3.19). With the results obtained through the load calculation and the manufacturer catalogue [17] it is selected a machine with a cooling capacity of 241 kW to cover the energy needs of the building (Table 3.17).

Table 3.17. Chiller data sheet

Chiller	
Model	AIRLAN NRL 0900
Refrigerant	R410
Cooling Capacity (kW)	241
Total power input (kW)	81,7
Heat recovery power (kW)	318
Fans	4 (Axial)
Dimensions (mm)	3400x2200x2450
Weight (kg)	2265
Compressors	Scroll

Condensator	
Fluid	Air
Inlet/Outlet Temperature(ºC)	33/46
Flow (m^3/h)	71500
Evaporator	
Fluid	Water
Inlet/Outlet Temperature(ºC)	7/12
Flow (m^3/h)	38700
Power (kW)	241
Pressure Drop (m.c.a)	6
Recovery System	
Fluid	Water
Inlet/Outlet Temperature(ºC)	50/44
Flow (m^3/h)	46000
Power (kW)	318
Pressure Drop (m.c.a)	7

In order to fulfill the RITE regulations IT1.2.4.1.3.1 and IT 1.2.4.1.3.2. [4] the energy efficiency ratio in cooling (EER) and the Seasonal Energy Efficiency Ratio (SEER) are critical.
Those ratios are important in order to maximize energy savings, the SEER rating of a unit is the cooling output during a typical cooling-season divided by the total electric energy input during the same period. The higher the unit's SEER rating the more energy efficient it is. The EER is the same but at a given operating point.

Table 3.18. EER in different operation modes

Outside Temperature (ºC)	Operation %	Time %	Cooling Capacity (kW)	Energy Input (kW)	EER
35	100	3	240,7	80,8	2,98
30	75	33	193,7	53,9	3,59
25	50	41	137,9	31,1	4,43
20	25	23	73,3	14,3	5,13

In the Table 3.18 are summarized the EER for different operation modes depending the load variation, the SEER results in 4,27 satisfying the requirements. This table with be useful later when calculating the amount of energy that is possible to recover, it estimates the time percentage that will be operating at different outside temperatures based in the temperature records of the city.

The Table 3.17 shows that the chiller is able to produce 318 kW to heat the water with a 100% performance and the water flow is 46000 m^3/h with a 44-50 temperature difference. It will be assumed the chiller operating at 225kW that is the cooling load calculated and the recovering system at 300kW.

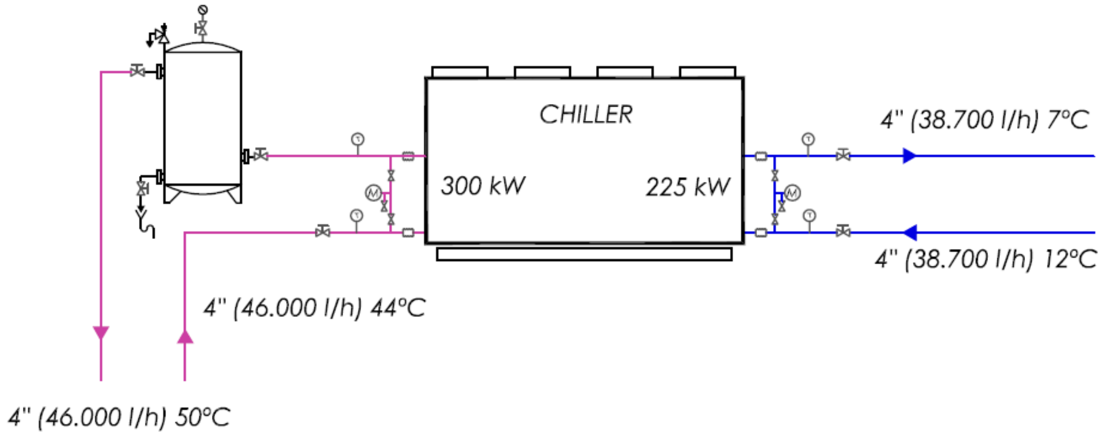

Figure 3.20. Chiller hydraulic system

In the Figure 3.20 is represented the hydraulic system with the appropriated vibration isolators, pressure gauges, thermometers, valves and a buffer tank to have available hot water in case the chiller is not working simultaneously. The pink pipelines will be connected to our system and the blue ones are responsible for the AHU-units or fancoils of the building.

3.8 Boiler

A boiler to heat the water of the pool the first time at the start-up system and supply de AHU-units and fancoils is required. A Condensing boiler with high efficiency, which uses the waste heat in the flue gases pre-heating the inlet cold water is selected. The difference with a conventional boiler is that this kind of boiler extracts additional heat from the waste gases by condensing the water vapor, produce during the combustion, to liquid, leaving the system via a drain, thus recovering its latent heat of vaporization.

Tabla 3.19. Power heating summary

	Pools Water	Pools Air	Fancoils/AHU-units	TOTAL
Power required (kW)	101	92	170	363

Considering the heating of the pools air and water and the building heating load, the total power needed by the system is 363 kW, therefore a condensing boiler of 450 kW is selected from Ygnis catalogue [18], which has models from 40 kW to 600 kW (Table 3.20). The pools water and air heating is considered because of the system start-up and in case there were a failure. The dehumidifiers has an extra coil to heat the air if it is necessary using water from the boiler.

Table 3.20. Condensing boiler technical sheet

CHARACTERISTICS									
Models			116	145	180	220	270	330	450
Boiler output		kW	116	145	180	215	270	330	450
Thermal efficiency (ncv) FULL LOAD	80/60°C	%	95.7	94.5	94.8	92.3	94.1	92.1	95.3
	40/30°C	%	104.0	102.8	102.3	99.3	100.2	100	99
Thermal efficiency (gcv) FULL LOAD	80/60°C	%	86.1	85.1	85.3	83.1	84.7	85.7	85.9
	40/30°C	%	93.6	92.5	92.1	89.4	90.2	90.1	89.2
Nominal gas inlet pressure		mbar	20						
Minimal gas inlet pressure		mbar	17.5						
Maximum gas inlet pressure		mbar	25						
Gas consumption, G20 (mini)		m³/h	3.33	4.13	5.08	9.39	11.66	9.31	12.49
Gas consumption, G20 (max)		m³/h	12.9	16.4	20.3	25.4	31.1	36.9	50
Dry weight		kg	220	230	235	340	370	440	525
Water capacity (approx)		litre	66	70	74	90	115	162	190
Exhaust gas emission Temperature 80/60°C	Min load	°C	60	60	60	70	80	56	60
Exhaust gas emission Temperature 80/60°C	Max load	°C	110	115	120	160	160	130	130
Exhaust gas emission Temperature 40/30°C	Min load	°C	35	35	35	40	45	33	37
Exhaust gas emission Temperature 40/30°C	Max load	°C	70	75	90	135	135	98	90
Nox emission		mg/m³	<50	<50	<50	<60	<60	<60	<50
C0 emission		mg/m³	<35						
Carbon intensities		kg/kWh	<0.063						
Minimum water flow rate at min. output		l/sec	0.35	0.43	0.54	1.03	1.29	0.99	1.35
Minimum water flow rate at max. output		l/sec	1.39	1.73	2.15	2.57	3.23	3.94	5.39
Maximum water flow rate		l/sec	2.77	3.46	4.30	5.26	6.45	7.89	10.78
Hydraulic resistance main exchanger	Delta T 20K	mbar	85	105	135	97	140	92	148
Hydraulic resistance main exchanger	Delta T 10K	mbar	340	420	540	388	560	368	592
Hydraulic working pressure	min	bar	1. cold			6			
	max	bar	4			6			
Test pressure		bar	6			9			
Flow water temperature range		°C	up to 85 °C						
Boiler overheat thermostat setting		°C	110 °C						
Minimum return water temperature		°C	NO RESTRICTION						
Electricity supply			230V 1 PH. 16A + neutral + ground 5C						

Assuming a 50-40,5 ºC temperature difference in the water flow for the AHU-units, 50-38 ºC for the fancoils and 50-42 ºC for the pool water heating pipeline the water flow is determined. In the Figure 3.21 is shown the boiler connected to a separator manifold dimensioned to achieve a very low water flow rate in order to avoid interference between the circulation pumps of the different loops, all the system with the appropriated pressure gauges, thermometers, valves, collectors and pumps.

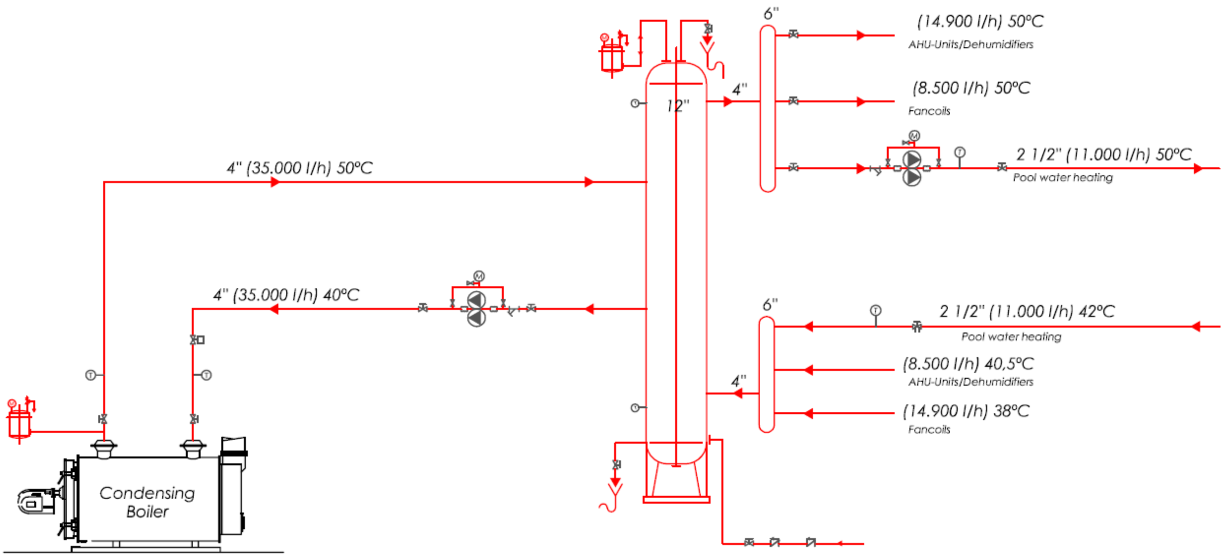

Figure 3.21. Condensing Boiler hydraulic diagram

3.9 Pumps

Centrifugal pumps are used in HVAC systems for circulation of chilled, hot, and condensing water in which the rotating action of the impeller generates a pressure which forces the fluid through the pipelines.

In order to size the pumps of the system it is necessary to define:

- The flow rate of liquid
- The liquid density
- The system pressure drop at the design flow rate

Losses include pipe, fittings, valves, and equipment such as heat exchangers, boilers, or chillers.

A software developed by Ebara [19], a pumps manufacturer, will be used to choose the proper design operating point in the performance pump curve, analyzing different pump curves to find the best efficiency and lowest power. The program calculates the pressure drop in the line considering the length, diameter, valves, filters, fittings, elbows and equipment. It is necessary to introduce the diameter of each valve and filter as well as the equipment pressure drops supply by the manufacturers catalogues [19].

Table 3.21. Data input for pumps selection

Line	Flow rate (l/h)	Length (m)	Valves	Elbows	Filters	Equipment
Boiler	35000	30	5	3	1	Boiler
Chiller	46000	70	6	20	1	Chiller
Pool water heating	9500	170	8	24	1	3 Heat Exchangers
Pool water heating	11000	160	11	20	1	Heat Exchanger
Pool for activities	9400	30	8	12	1	Heat Exchanger Dehumidifier
Swimming pool	23500	30	8	12	1	Heat Exchanger Dehumidifier

The table 3.22 summarize the pumps used in the system ant their characteristics. The pump performance curves are presented in the appendix A.

Table 3.22. Pump selection summary

	Model	Capacity (m^3/h)	Pressure drop (kPa)	Motor power (kW)	Nominal Speed (r.p.m)	Frecuency (Hz)	Discharge Diammeter (mm)
Pump 1	ELD 80-160	46	78	1,5	1450	50	116
Pump 2	ELD65-160	35	59	1,1	1450	50	116
Pump 3	ELD 50-250	9,5	186	2,2	1450	50	116
Pump 4	EL 50-200	9,4	107	1,1	1450	50	116
Pump 5	El 65-200	23,5	117	1,5	1450	50	116
Pump 6	ELD 50-250	11	215	2,2	1450	50	116

3.10 Heat Exchangers

The plate heat exchangers used in our system are sized using a software developed by Sedical, a Spanish manufacturer and its sizing guide [21]. Sedical has a wide range of plate exchangers designed for industrial and HVAC applications.

Figure 3.22 Sedical Heat Exchanger

Table 3.23. Heat Exchanger 1 data

	Heat Exchanger 1	
OPERATION	**Normal**	**Start-Up**
FLOW 1	11.000 l/h	11.000 l/h
IN/OUT TEMPERATURE	50/42 ºC	77,5/51 ºC
FLOW 2	11.000 l/h	11.000 l/h
IN/OUT TEMPERATURE	37/29 ºC	50/23,5 ºC
POWER	101 kW	338 kW
MAX. PRESSURE DROP	5 mca	
MODEL	UFP-54 / 15 M - C1 - PN10	

Table 3.24. Heat Exchanger 2 Data

	Heat Exchanger 2	
OPERATION	**Normal**	**Start-Up**
FLOW 1	7.300 l/h	7.300 l/h
IN/OUT TEMPERATURE	50/42 ºC	78/50 ºC
FLOW 2	23.500 l/h	23.500 l/h
IN/OUT TEMPERATURE	31,5/29 ºC	38/29 ºC
POWER	68 kW	240 kW
MAX. PRESSURE DROP	5 mca	
MODEL	UFP-52 / 29 LH 43 - C - PN10	

Table 3.25. Heat Exchanger 3 Data

	Heat Exchanger 3	
OPERATION	**Normal**	**Start-Up**
FLOW 1	2.200 l/h	2.200 l/h
IN/OUT TEMPERATURE	50/42 ºC	78/56 ºC
FLOW 2	9.400 l/h	9.400 l/h
IN/OUT TEMPERATURE	32/30 ºC	35/30 ºC
POWER	20 kW	57 kW
MAX. PRESSURE DROP	5 mca	
MODEL	UFP-32 / 30 H - C - PN10	

In the tables 3.23, 3.24 and 3.25 are disposed the Heat exchangers characteristics designed based in the inlet/outlet temperature and the flow rate in the primary and secondary lines.

3.11 Final System Diagram

The system diagram with all the dimensions is shown in Figure 3.23. The chiller is located on the roof, where the recovery system pipelines go down through a shaft to the heat exchangers placed in the basement, close to the dehumidifiers. The heat recovered is expected to contribute the DHW system used in the sport center, so an outlet and inlet pipeline is done in advance.

The water pipelines from the pool are connected to the dehumidifier and the heat exchanger from the chill recovery system and the boiler is connected to that line with another heat exchanger.

All the shutoff, balancing and two or three way valves are disposed to make easier the dismantling and regulation in the network. The twin pumps are made of the same basic components and hydraulic design as the single In-line pumps, but are comprises of two pumps in one common casing incorporated with a non-return valve which prevents the internal circulation in case only one pump is running or the other is dismounted for repair.

Pressure gauges and thermometers are included to control the correct performance of the system.

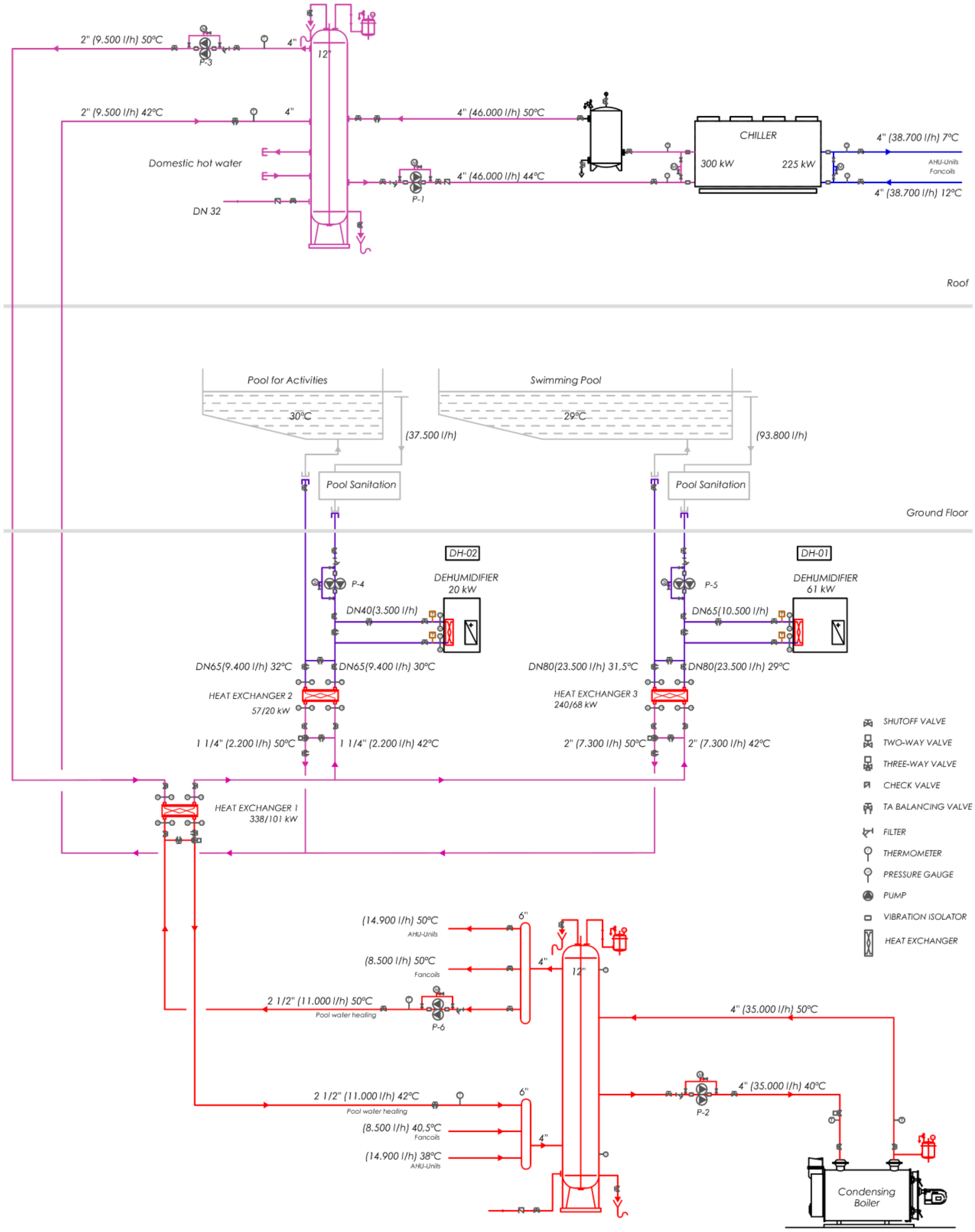

Figure 3.23. Final system diagram

4. COMPARATIVE WITH A SOLAR SYSTEM

The minimum solar contribution to heat swimming pools water is regulated by the RITE IT 1.2.4.6.2 [4] which specifies that thermal installations for those purpose shall accomplish with the requirement set in the HE 4 "Minimum solar contribution to domestic or pools hot water" section of the Technical Building Code.

According to the document, at least a 50% solar contribution of the annual energy demand is required for heating the water. Therefore, a calculation with solar panels is performed to cover 50% of the demand in Appendix B.

The result of the calculation for that specification, shows an annual output of 507891 kWh produced by solar collectors, with an annual heat demand of 1011610 kWh.
The RITE in its IT.1.2.2 instruction [4] establish the energy efficiency requirements. It indicates that is possible to design and size heating alternative systems on condition that the energy effiency is improved with appropriate proofs.

Since the system has two heat recovery sources, one from the condensation coil of the dehumidifiers and other from the exhaust air from the chiller, it is proved that those sources cover the energy that would be obtained with the solar collectors required. The design contemplates, at first instance, heating the pool water by heat exchange with the energy produce in the condensing coil in the dehumidifiers units and secondly, the heating by heat exchange with the exhaust air from the chiller.

The energy input obtained from these systems is quite superior to that obtained from the solar collectors, so that, according to the regulation, the performance of the system designed satisfies the technical requirements.

In the Table 4.1 it is shown the energy produced by the dehumidification equipment operating in a regular situation contributing with a 55/45% of the heat recover to the air and water.

Table 4.1. Energy produced by the dehumidifier condensing coils

	Air	Water
Swimming Pool	67,7 kW	61 kW
Pool for Activities	26,4 kW	19,6 kW
Total	175kW	

The operating conditions are equivalent to 15 hours per day, 365 days a year. The demand for the dehumidification equipment in those conditions is as follows:

- Three months per year, the units are operating at 25% of the capacity and using free-cooling mode, corresponding to 1380 hours / year.
- Four months per year, the units are operating at 50% of the capacity which is equivalent to 1,830 hours / year.
- Two months per year, the units are operating at 75% which is equivalent to 915 h / year.
- The units are operating at full capacity three months per year which is equivalent to 1350 hours / year.

The operation is simulated monthly and the energy recovered is calculated in table 4.2.

Table 4.2. Energy recovered by the dehumidifiers

Month	Operation (%)	Hours/day	Days/month	Hours/month	Heating Effect (kW)	Heating Energy (kWh)
January	100	15	31	465	175	81236
February	100	15	28	420	175	73374
March	75	15	31	465	131	60927
April	50	15	30	450	87	39308
May	50	15	31	465	87	40618
June	25	15	30	450	44	19654
July	25	15	31	465	44	20309
August	25	15	31	465	44	20309
September	50	15	30	450	87	39308
October	50	15	31	465	87	40618
November	75	15	30	450	131	58961
December	100	15	31	465	175	81236
						575855

The energy recovered from the dehumidification equipment is 575855 kWh, higher than the 507891 kWh annually obtained by the solar system.

The chiller operating conditions, respond to 12 hours per day, 365 days a year. The demand for cooling in these conditions according to the coefficients SEER (Table 3.18) is represented in Table 4.3.

Table 4.3. Energy recovered by the chiller

Month	Operation (%)	Hours/day	Days/month	Hours/month	Heating Effect (kW)	Heating Energy (kWh)
January	0	12	31	372	0	0
February	0	12	28	336	0	0
March	0	12	31	372	0	0
April	0	12	17	204	0	0
	25	12	13	156	69	10725
May	25	12	17	204	69	14025
	50	12	14	168	138	23100
June	50	12	21	252	138	34650
	75	12	9	108	206	22275
July	75	12	31	372	206	76725
August	100	12	6	72	275	19800
	75	12	25	300	206	61875
September	50	12	30	360	138	49500
October	50	12	16	192	138	26400
	25	12	15	180	69	12375
November	0	12	30	360	0	0
December	0	12	31	372	0	0
						351450

The energy from the chiller recovery system is 351450 kWh that in addition to the energy recover from the dehumidifiers provides 927305 kWh per year, significantly higher than the 507891 kWh required by regulations. Although the chiller recovery system is intended to supply a possible domestic hot water system, this minimum contribution is already covered only with the dehumidifiers, which provide 575855 kWh.

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5. CONCLUSION

Along this project, it has been studied the conditioning of the air and the heating of the water in a pools facility in a sport center with energy recovery systems.

The study was conducted based on the dehumidification needs and the energy loss through the enclosure, sizing two dehumidifiers to condition the air. The two dehumidifiers are responsible of reduce the humidity in the air cooling it firstly and heating it again to adapt it to the temperature in the room. Those units are capable of recover energy by means of a second coil that heat the water recirculated in the pool.

A load calculation for the entire sport center building was performed in order to size a chiller and use the heat produced in the evaporator coil through a recovery system added to the equipment, to heat the pool water. This calculation was done using HAP software to simulate the most accurate possible situation.

The system consists of two dehumidifiers, a chiller recovery system, a boiler, heat exchangers, pumps and the hydraulic components needed to assure a proper operation.

The system is developed aiming to find a more efficient way to heat the pools water since following Spanish Regulations [4], is mandatory to cover a 50% of the annual demand of the energy needed to heat the water with a solar energy system.
There is an exception to use other systems in case is justified the use of a more efficient way.

One of the most notable points when using this type of systems is that the chiller is needed to supply the AHU-units and fancoils in the building, so it does not involve high installation costs.

Using software, is estimated that 248 solar collectors are needed to cover the demand and the output energy obtained from them results in 507891 kWh. The collector specifications are shown in Appendix B.
The energy recovered from the dehumidification equipment is 575855 kW, higher than the annual energy obtained by the solar system. In addition, the chiller recovery system produces 351450 kWh.

Finally, as it is known solar systems have great environmental savings, they could be great options in houses or residential buildings where there is no possibility to install other systems as chillers or AHU units, but as it has been demonstrated, there are better options to improve the efficiency and reduce the environmental impact in bigger facilities as sport centres. Therefore, it is not always recommended to apply standards or regulations as the RITE [4] without seeking more efficient methods to resolve our system problems.

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APENDIX A: PUMP PERFORMANCE CURVES

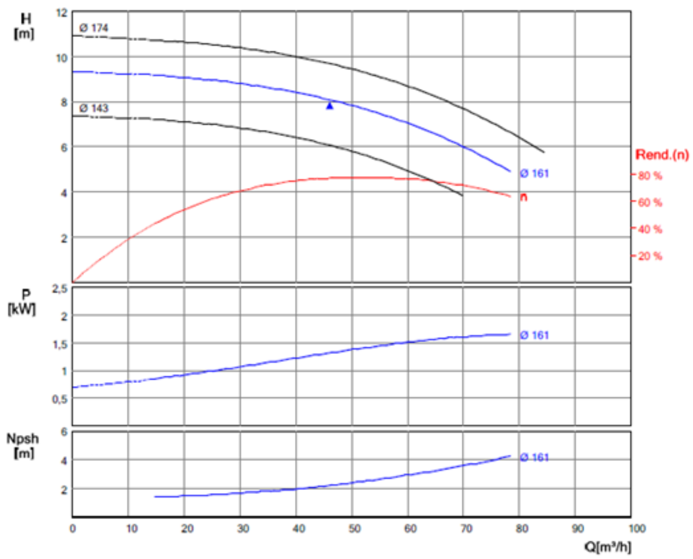
EBARA

EBARA ESPAÑA BOMBAS, S.A.
Pol.La Estación. C/Cormoranes,6
Tel.916 923 630, Fax 916 910 818
28320 Pinto(Madrid), ESPAÑA
http://www.ebara.es

GRUPO MOTOBOMBA

· Modelo : **ELD 80-160**
· Motor : **1450-1,5 kW**
· Fluido : Agua dulce, limpia, temperatura ambiente
· Tensión : 400V III+N, 50Hz

PUMP 1



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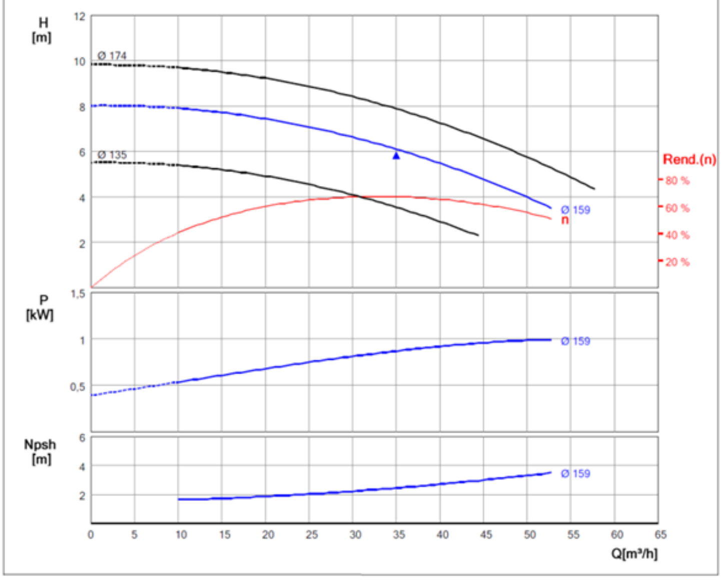
Pol.La Estación. C/Cormoranes,6
Tel.916 923 630, Fax 916 910 818
28320 Pinto(Madrid), ESPAÑA
http://www.ebara.es

ELD 65-160
1450-1,1 kW
400V III+N, 50Hz

PUMP 2



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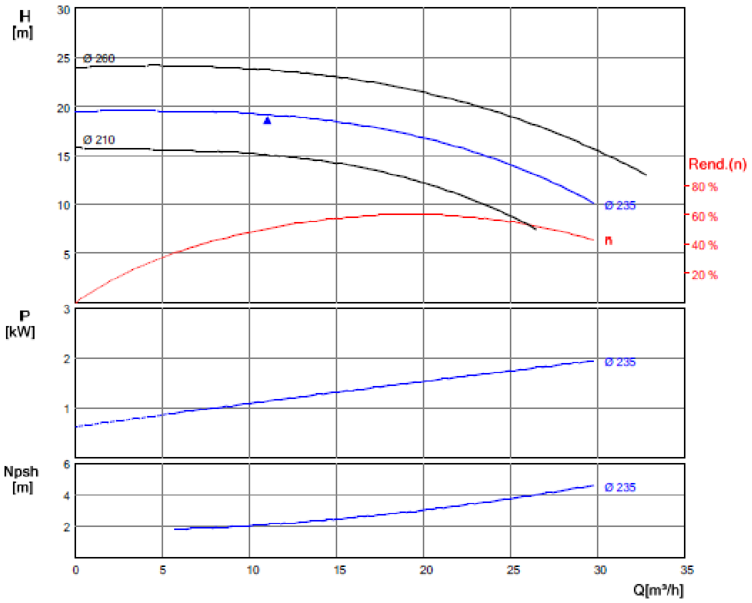
Pol.La Estación. C/Cormoranes,6
Tel.916 923 630, Fax 916 910 818
28320 Pinto(Madrid), ESPAÑA
http://www.ebara.es

GRUPO MOTOBOMBA
· Modelo : **ELD 50-250**
· Motor : **1450-2,2 kW**
· Fluido : Agua dulce, limpia, temperatura ambiente
· Tensión : 400V III+N, 50Hz

PUMP 3



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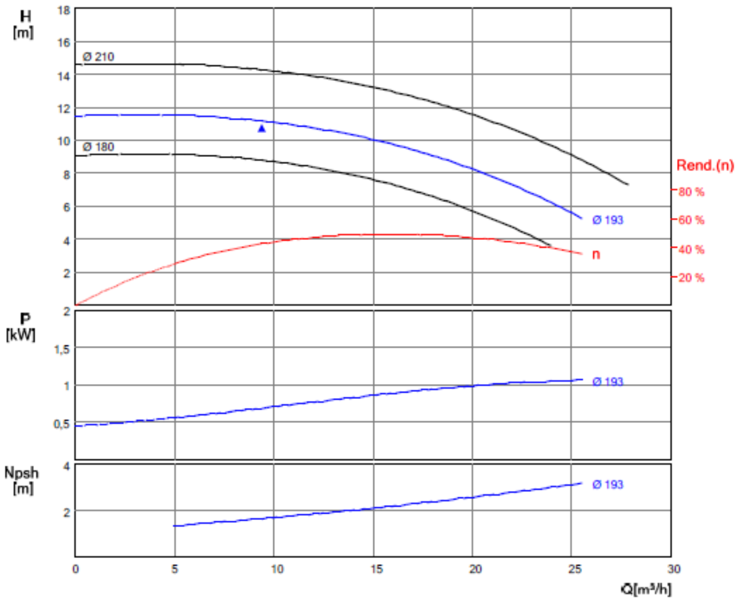
Pol.La Estación. C/Cormoranes,6
Tel.916 923 630, Fax 916 910 818
28320 Pinto(Madrid), ESPAÑA
http://www.ebara.es

GRUPO MOTOBOMBA
· Modelo : **EL 50-200**
· Motor : **1450-1,1 kW**
· Fluido : Agua dulce, limpia, temperatura ambiente
· Tensión : 400V III+N, 50Hz

PUMP 4



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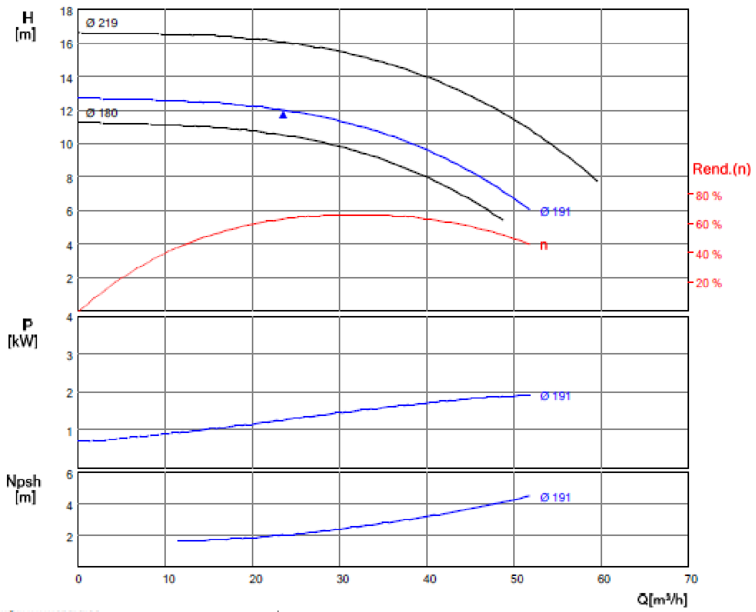
Pol.La Estación. C/Cormoranes,6
Tel.916 923 630, Fax 916 910 818
28320 Pinto(Madrid), ESPAÑA
http://www.ebara.es

GRUPO MOTOBOMBA
· Modelo : **EL 65-200**
· Motor : **1450-1,5 kW**
· Fluido : Agua dulce, limpia, temperatura ambiente
· Tensión : 400V III+N, 50Hz

PUMP 5



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Pol.La Estación. C/Cormoranes,6
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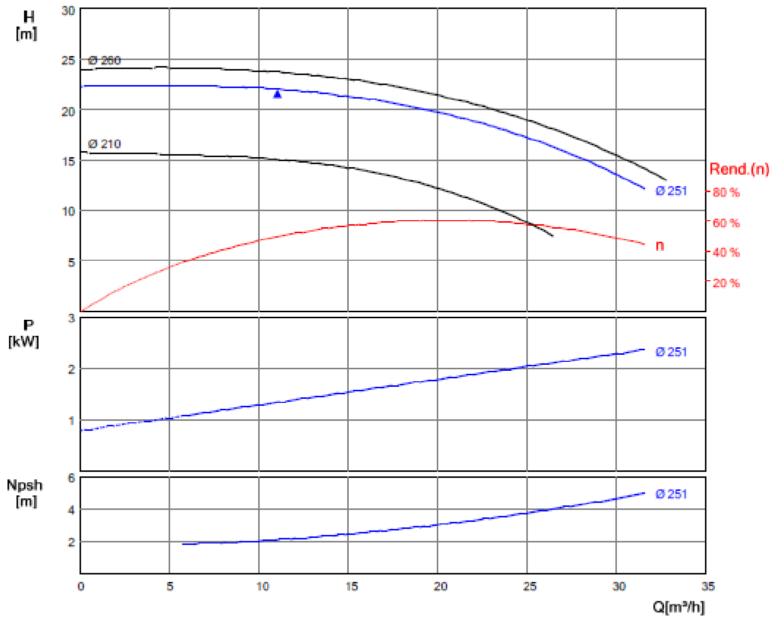
28320 Pinto(Madrid), ESPAÑA
http://www.ebara.es

GRUPO MOTOBOMBA
· Modelo : **ELD 50-250**
· Motor : **1450-2,2 kW**
· Fluido : Agua dulce, limpia, temperatura ambiente
· Tensión : 400V III+N, 50Hz

PUMP 6



APPENDIX B. SOLAR COLLECTORS SYSTEM

Sedical S.A. - WEISHAUPT Solar collectors

System description:

Climate Data

SEGOVIA					
Altitude (m)	1002	**Latitude (º)**	41		
	Average Temperature ºC	**Relative Humidity %**	**Average water Temperature ºC**	**Number of Hours of sun**	**Average Solar Radiation in an horizontal surface kWh/m^2**
January	4	76	4	248	49.08
February	6	72	5	252	68.44
March	10	65	7	279	115.39
April	12	66	9	285	153.33
May	15	63	10	294	175.67
June	20	57	11	285	188.33
July	24	49	12	294	221.31
August	23	50	11	294	214.42
September	20	57	10	270	156.67
October	14	68	9	279	98.17
November	9	75	7	240	56.67
December	5	78	4	232	43.92

Pool Data

Pool	Indoor	**Enclosure Temperature**	28	ºC
Number	2	**Enclosure Relative Humidity**	65	%

	Occupancy rate (%)	**Number of hours per day**
January	100	15
February	100	15
March	100	15
April	100	15
May	100	15
June	100	15
July	100	15
August	100	15
September	100	15
October	100	15
November	100	15
December	100	15

		Pool 1	**Pool 2**	**Pool 3**
Thermal Pool cover		No	No	--
Shape factor	%	10	10	--
Area	m^2	312,5	75	--
Width	m	12,5	6	--
Volume	m^3	375	75	--
Water Temperature	ºC	29	30	--
Occupancy rate	Swimmers / h m^2	0.15	0.15	--

Sedical S.A. - WEISHAUPT Solar collectors

Efficiency Curve

$R = Ro - a1\,x - a2\,I\,x^2$

$Ro = 0.814$

$a1 = 3.527\ W/m^2K$

$a2 = 0.012\ W/m^2K^2$

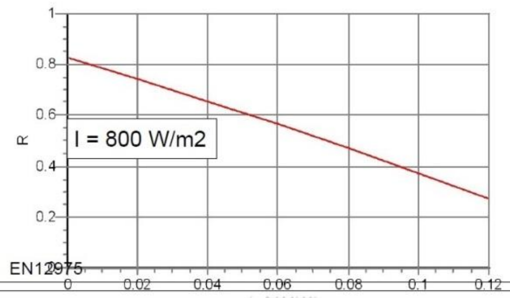
$I =$ Average Solar Radiation W/m^2

$x = (tm - ta)/I \quad (m^2K/W)$

Collectors layout

Inclination (º)	45
Azimuth (º)	-45

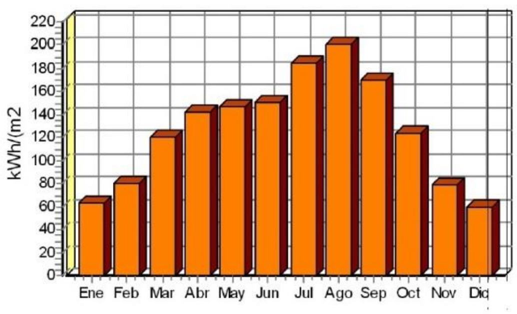
Efficiency Curve — WEISHAUPT WTS-F2 K3/K4



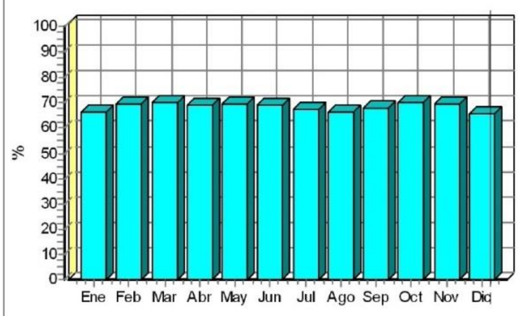
Energy Balance — WEISHAUPT WTS-F2 K3/K4

	Monthly energy gross gain kWh/m²	Average Solar radiation W/ m²	Average efficiency %	Monthly energy net gain kWh/m²
January	62.7	252.9	66.2	35.3
February	80.1	317.8	69.6	47.4
March	120.5	431.8	70.3	71.9
April	142.1	498.7	69.3	83.7
May	147.0	499.3	69.4	86.7
June	150.9	529.3	69.1	88.6
July	185.2	629.0	67.4	106.1
August	200.7	681.5	66.4	113.3
September	170.6	631.9	68.1	98.7
October	124.6	446.5	70.3	74.5
November	79.1	329.4	69.4	46.7
December	60.1	258.4	65.9	33.6

Monthly energy gross gain



Average collector efficiency



Sedical S.A. - WEISHAUPT Solar Collectors

Number of collectors	WEISHAUPT WTS-F2 K3/K4
	248

Absorber Area m^2
572.88

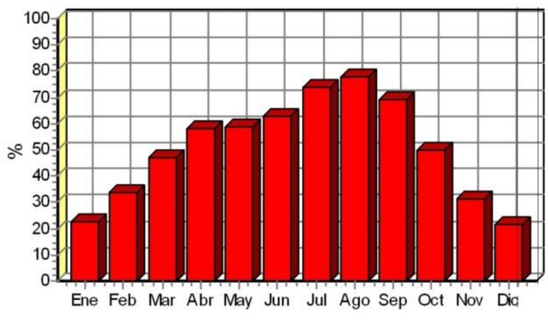
Collectors layout	
Minimun distance between horizontal collectors	289.0 cm
Minimun distance between vertical collectors	454.0 cm

Gross Area m^2
622.48

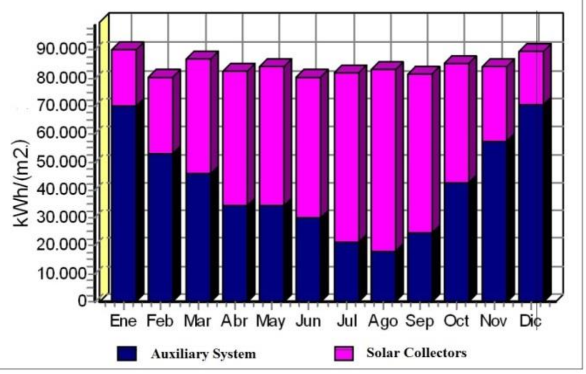
Energy Production WEISHAUPT WTS-F2 K3/K4

	Monthly Energy Demand kWh	Monthly net energy gain kW/h m^2	Monthly net energy produced by the collectors kW/h	Monthly coverage rate (%)
January	89988	35.3	20234	22.5
February	80434	47.4	27143	33.7
March	87171	71.9	41215	47.3
April	82624	83.7	47961	58.0
May	84385	86.7	49672	58.9
June	80599	88.6	50782	63.0
July	82226	106.1	60762	73.9
August	83130	113.3	64922	78.1
September	81422	98.7	56552	69.5
October	85283	74.5	42651	50.0
November	84404	46.7	26732	31.7
December	89945	33.6	19266	21.4
Anual	1011610		507891	50.21

Coverage rate



Energy Production

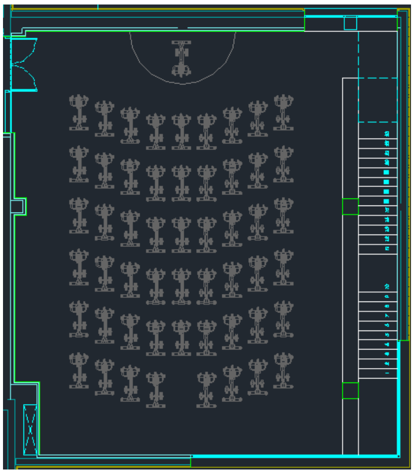


APPENDIX C. SPACE DISTRIBUTION FOR LOAD CALCULATION

Table. Zone 1 data and assumptions

Zone	Activities Room	
Floor Area	158 m²	
Avg. Ceiling Height	3,7 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	120 Watts	
Electrical Equipment Loads	3000 Watts	
Occupancy	53 People	
Sensible heat/person	154 W/person	
Latent heat/person	271 W/person	
Inner Wall	30 m²	0,86 W/ m²K
Inner Wall	35,5 m²	3 W/ m²K
Roof	158 m²	

Table. Zone 2 data and assumptions

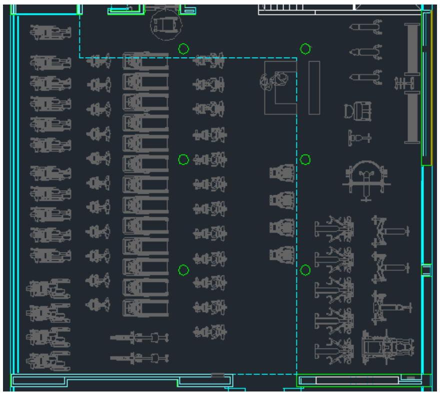


Zone	Spinning Room	
Floor Area	158 m²	
Avg. Ceiling Height	4,2 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	1500 Watts	
Electrical Equipment Loads	3000 Watts	
Occupancy	54 People	
Sensible heat/person	308 W/person	
Latent heat/person	435 W/person	
	m²	W/ m²K
NW Wall + Windows	52	3
SE Wall + Windows	52	3
SW Wall + Windows	60	3
Roof	158	

Table. Zone 3 data and assumptions

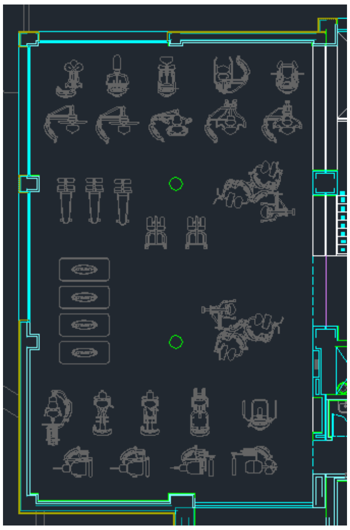
Zone	Activities Room	
Floor Area	81 m²	
Avg. Ceiling Height	3,7 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	720 Watts	
Electrical Equipment Loads	3000 Watts	
Occupancy	27 People	
Sensible heat/person	154 W/person	
Latent heat/person	271 W/person	
	m²	W/ m²K
Inner Wall	24	0,86
SE Wall + Windows	50	3
Roof	81	

Table. Zone 4 data and assumptions



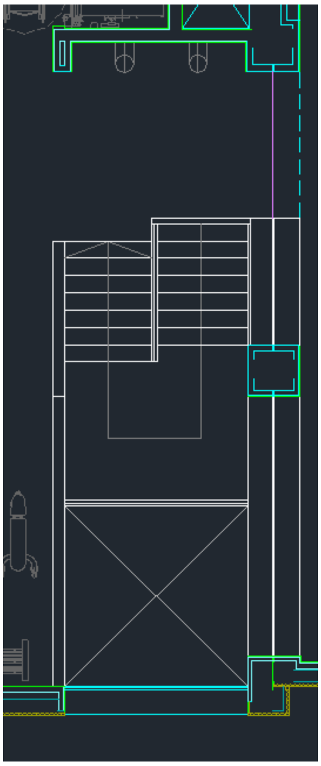
Zone	Fitness Room	
Floor Area	382 m²	
Avg. Ceiling Height	3,7 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	2000 Watts	
Electrical Equipment Loads	23400 Watts	
Occupancy	78 People	
Sensible heat/person	154 W/person	
Latent heat/person	271 W/person	
	m²	W/ m²K
SE Wall + Windows	98	3
Inner Wall	22	0,86
Inner Wall	40	3
Roof 1	232	
Roof 2	150	

Table. Zone 5 data and assumptions



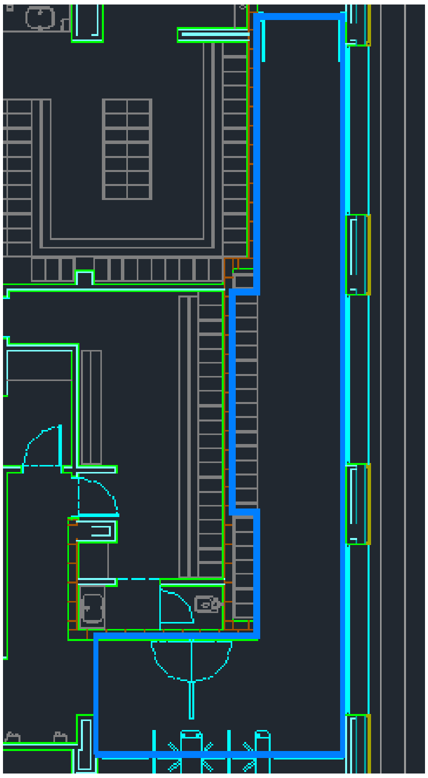
Zone	Fitness Room	
Floor Area	172 m²	
Avg. Ceiling Height	4,2 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	1440 Watts	
Electrical Equipment Loads	0 Watts	
Occupancy	30 People	
Sensible heat/person	154 W/person	
Latent heat/person	271 W/person	
	m²	W/ m²K
NW Wall + Windows	24	3
NE Wall + Windows	74	3
SE Wall + Windows	46	3
Inner Wall	8	0,86
Inner Wall	10	3
Roof	172	

Table. Zone 6 data and assumptions



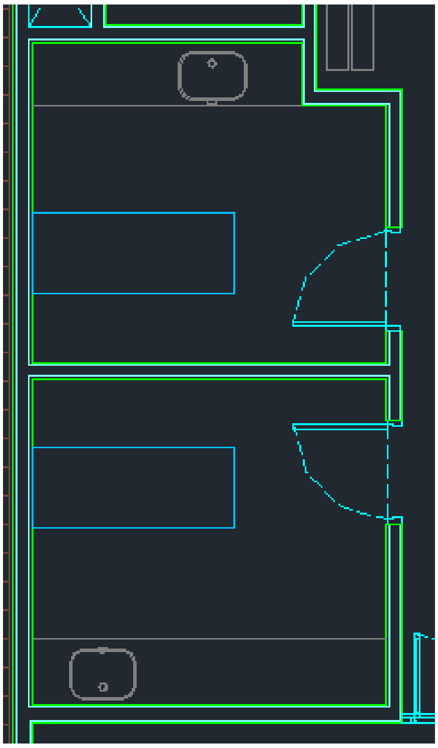
Zone		
Floor Area	41 m²	
Avg. Ceiling Height	4,2 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall + Windows	14	3
SW Wall	3,5	3
Inner Wall	20,2	0,86
Roof	172	

Table. Zone 7 data and assumptions



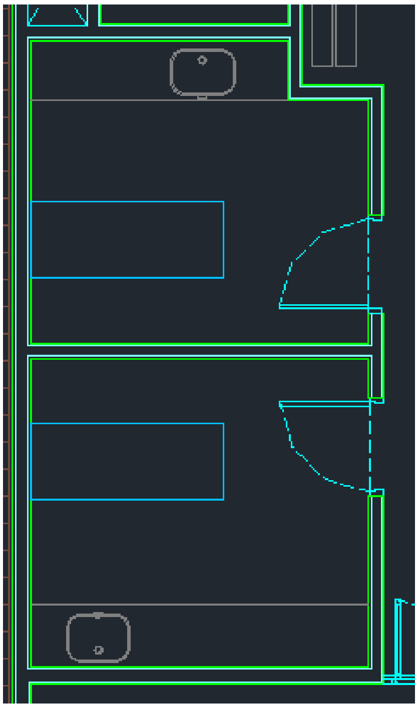
Zone		
Floor Area	38 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall + Windows	14	3
Inner Wall	28	0,86
Floor	38	0,640

Table. Zone 8 data and assumptions



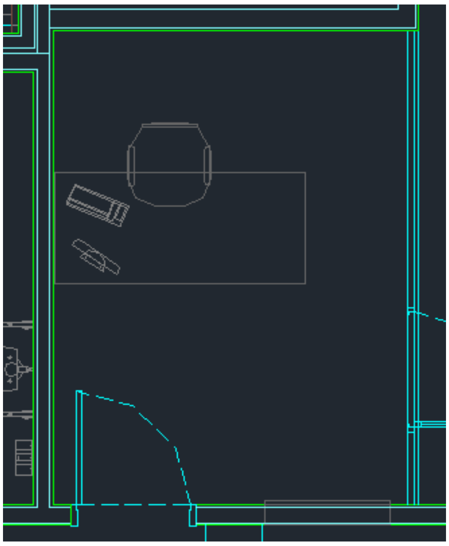
Zone	Massage Room	
Floor Area	9 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	2 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Inner Wall	8,2	0,86
Floor	9	0,640

Table. Zone 9 data and assumptions



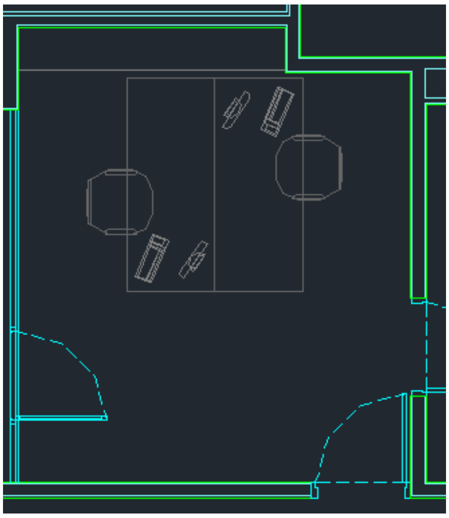
Zone	Massage Room	
Floor Area	9 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	2 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Inner Wall	8,2	0,86
Floor	9	0,640

Table. Zone 10 data and assumptions



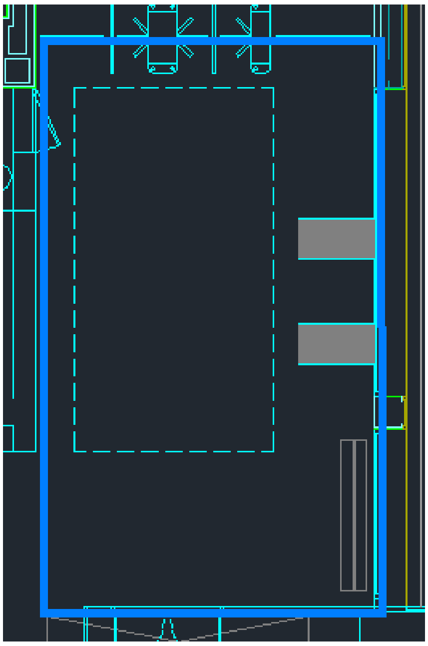
Zone	Management Room	
Floor Area	9 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	300 Watts	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Inner Wall	9	0,86
Floor	9	0,640

Table. Zone 11 data and assumptions



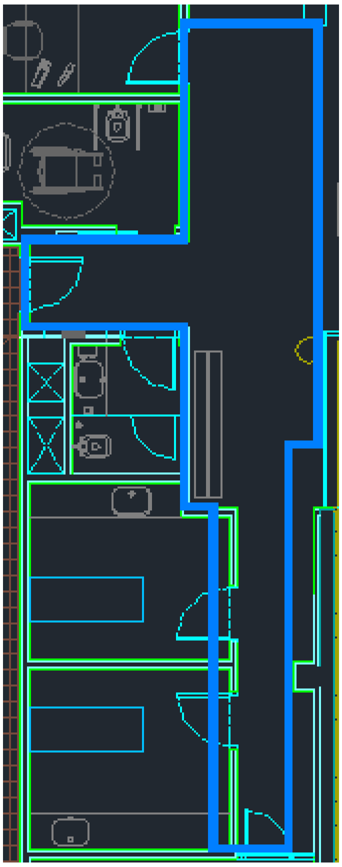
Zone	Administration Room	
Floor Area	14 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	600 Watts	
Occupancy	2 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Floor	14	0,640

Table. Zone 12 data and assumptions



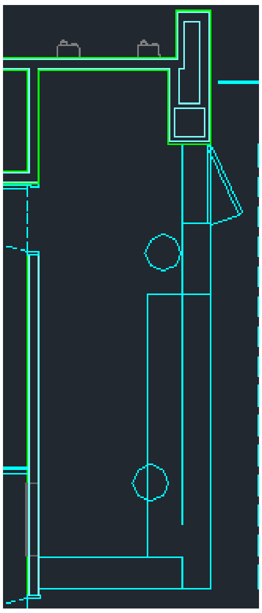
Zone	Entry	
Floor Area	46 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall + Windows	36	3
Floor	46	0,640

Table. Zone 12+ data and assumptions



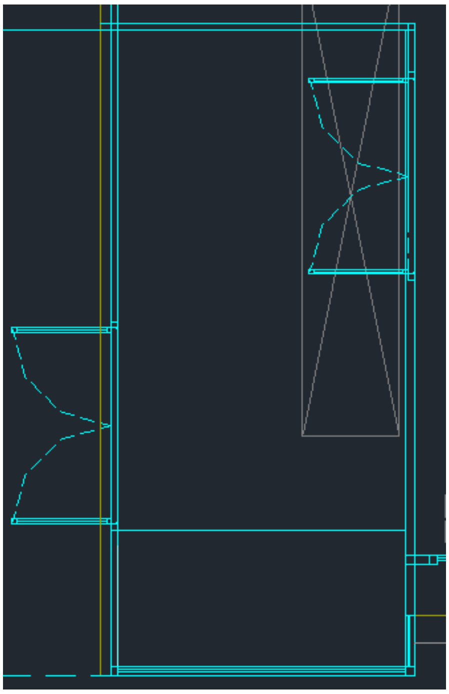
Zone		
Floor Area	28 m²	
Avg. Ceiling Height	4m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall + Windows	33,2	3
SW Wall + Windows	3,2	3
Inner Wall	24,4	0,86
Floor	28	0,640

Table. Zone 13 data and assumptions



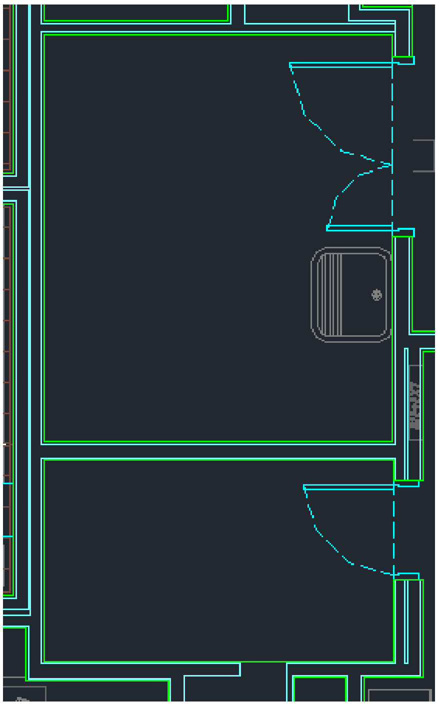
Zone	Reception Desk	
Floor Area	14 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	1000 Watts	
Occupancy	3 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Floor	14	0,640

Table. Zone 14 data and assumptions



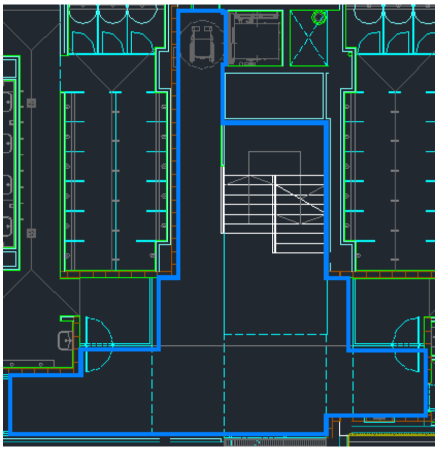
Zone	Entry	
Floor Area	11 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	30 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall + Windows	7	3
SW Wall + Windows	16	3
NE Wall + Windows	2	3
Floor	11	0,640

Table. Zone 15 data and assumptions



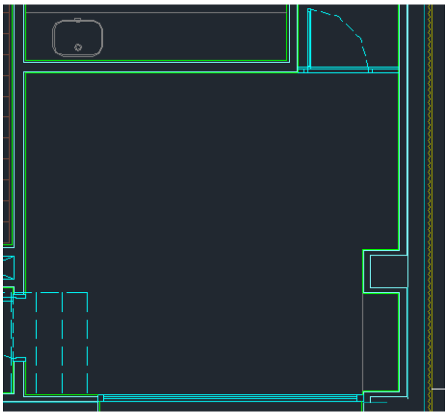
Zone	Swimming Instr.Room	
Floor Area	13 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Floor	13	0,640
Inner Wall	38	0,86

Table. Zone 16 data and assumptions



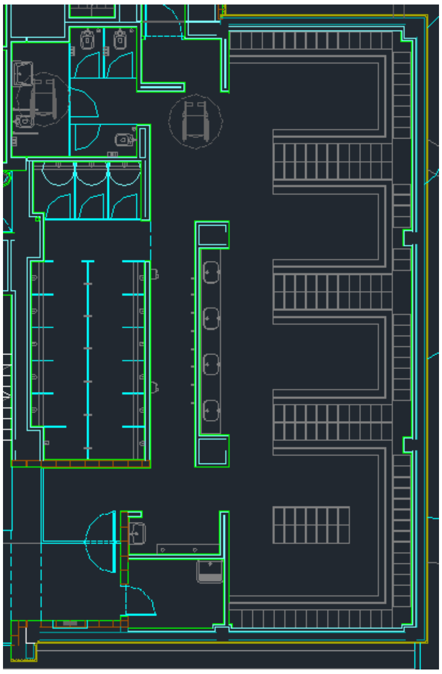
Zone		
Floor Area	67 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	0 Watts	
Occupancy	8 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall + Windows	27	3
Inner Wall	73,2	0,86
Floor	67	0,640

Table. Zone 17 data and assumptions



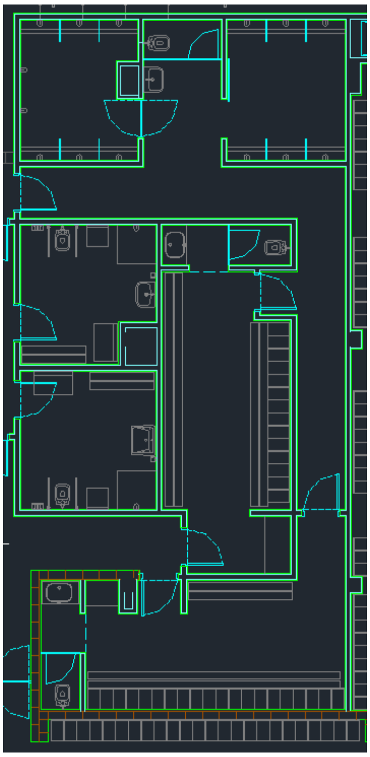
Zone		
Floor Area	17 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	0 L/s/person	
Lighting Loads	10 W/m²	
Electrical Equipment Loads	15 W/m²	
Occupancy	3 m²/person	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
SE Wall	19	3
SW Wall + Windows	16	3
NE Wall + Windows	2	3
Floor	17	0,640

Table. Zone 18 data and assumptions



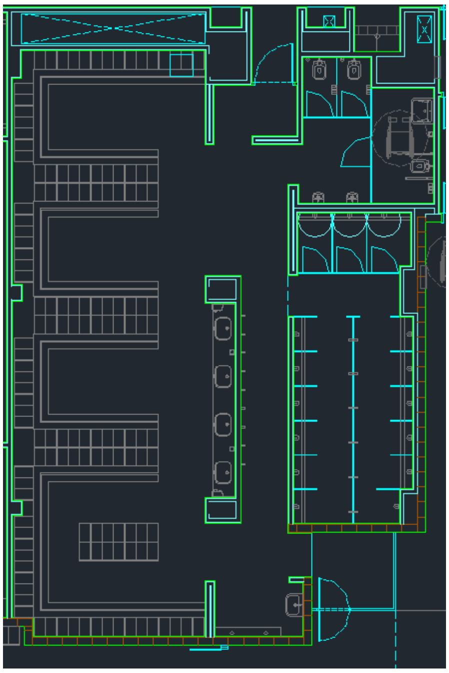
Zone	Locker Room	
Floor Area	127 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	2160 W	
Electrical Equipment Loads	6000 Watts	
Occupancy	60 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
NW Wall	22,5	3
NE Wall	70	3
SE Wall	22,5	3
Inner Wall	108	0,86
Floor	127	0,640

Table. Zone 20 data and assumptions



Zone	Locker Room	
Floor Area	78 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	1500 W	
Electrical Equipment Loads	6000 W	
Occupancy	20 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Inner Wall	2	0,86
Floor	78	0,640

Table. Zone 21 data and assumptions



Zone	Locker Room	
Floor Area	120 m²	
Avg. Ceiling Height	4 m	
Building Weight	341,8kg/m²	
Air Flow Rate:	8 L/s/person	
Lighting Loads	2160 W	
Electrical Equipment Loads	6000 W	
Occupancy	60 People	
Sensible heat/person	71,8 W/person	
Latent heat/person	60,1 W/person	
	m²	W/ m²K
Inner Wall	96	0,86
Floor	120	0,640

APPENDIX D. DIFFUSION STUDY CHARTS

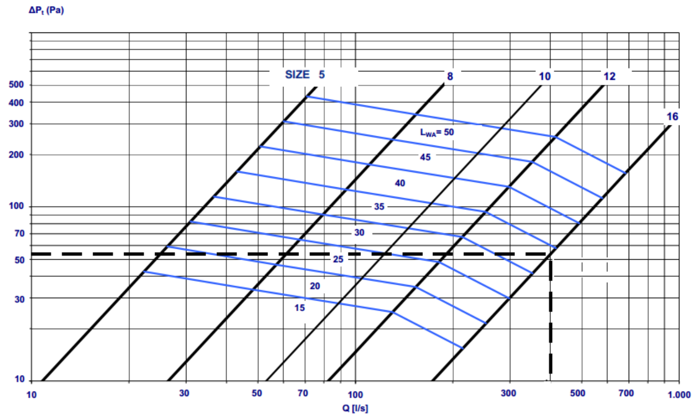


Figure 1. Pressure drop and sound power level

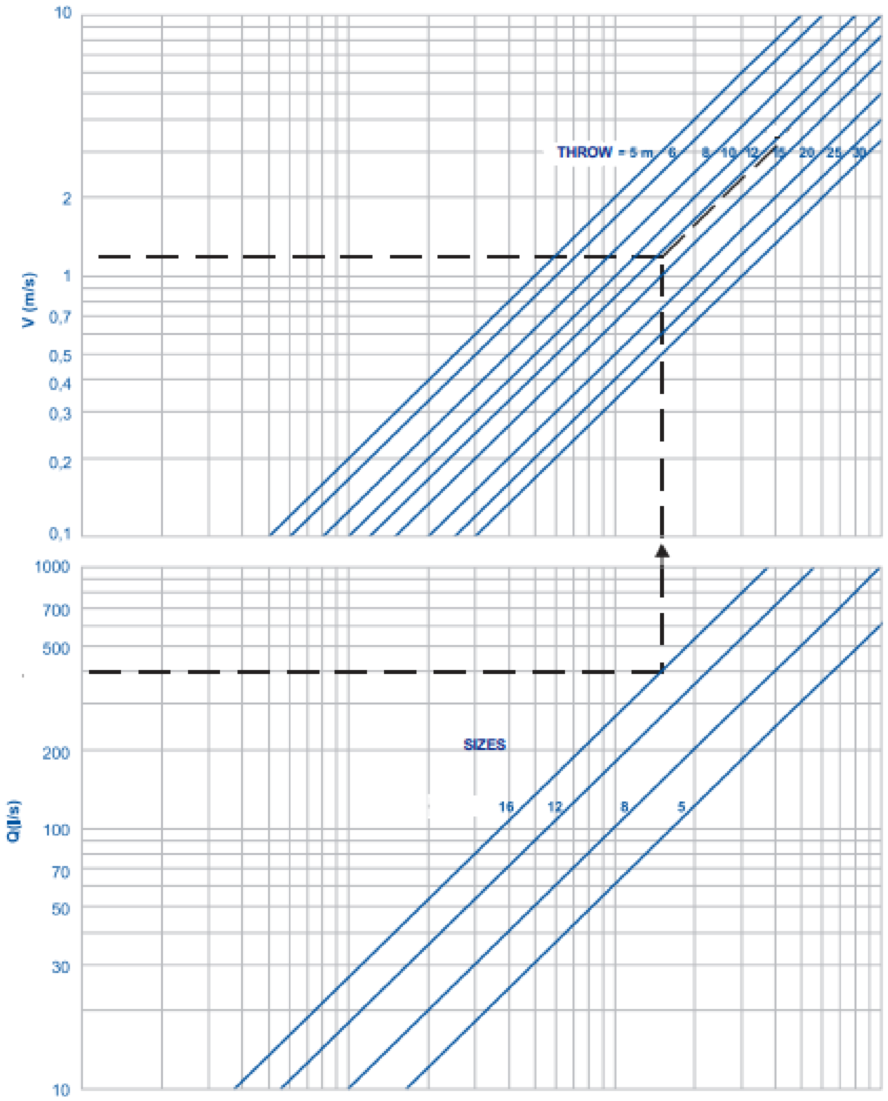


Figure 2. Velocity of the air jet for the throw

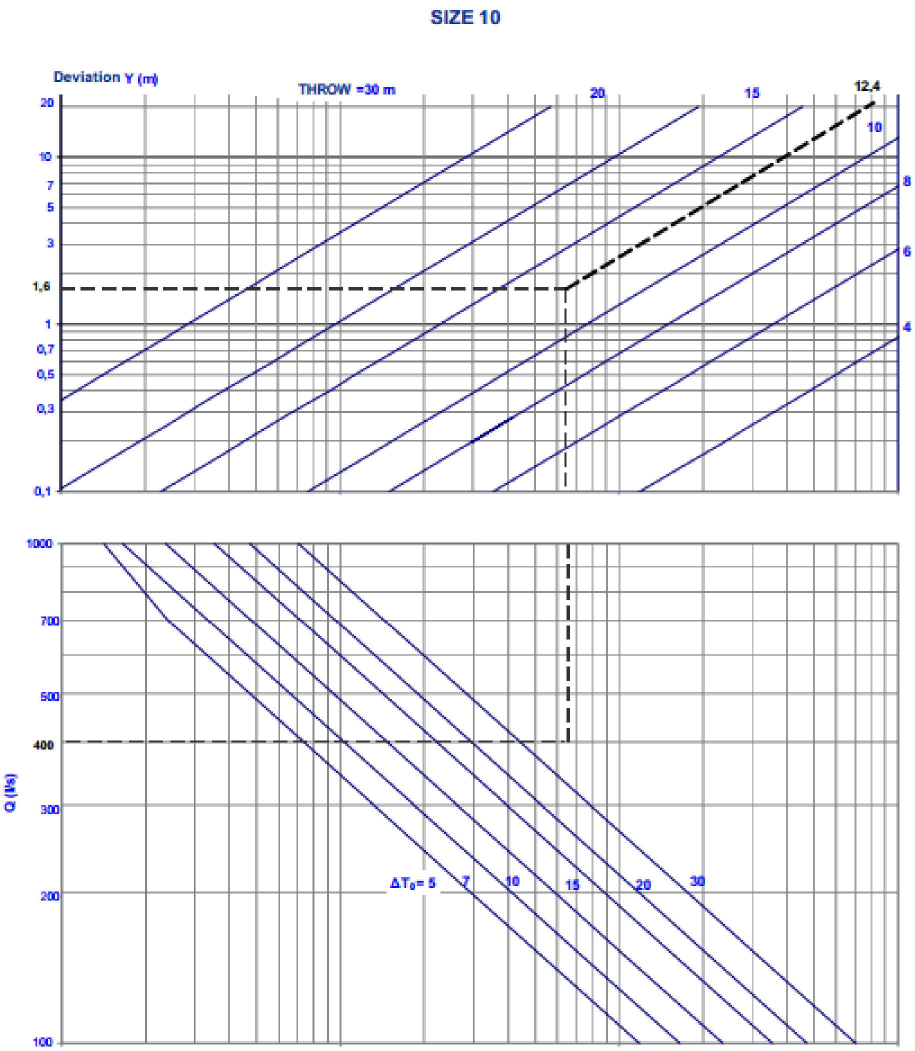


Figure 3. Vertical deviation of the air jet (non-isothermal jets)

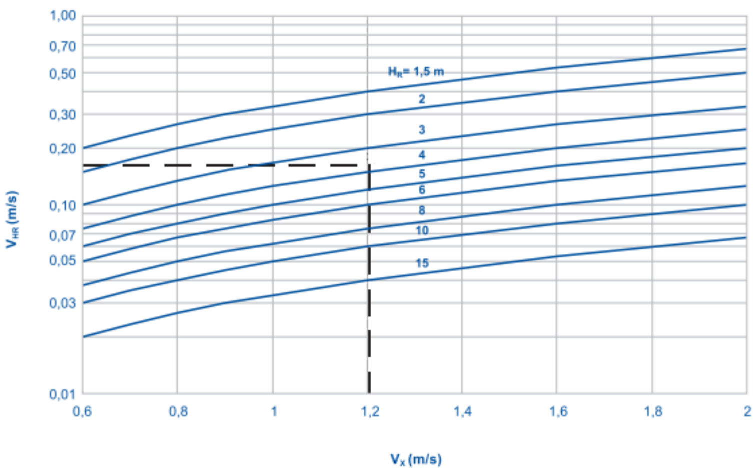


Figure 4. Ratio between air flow velocities.

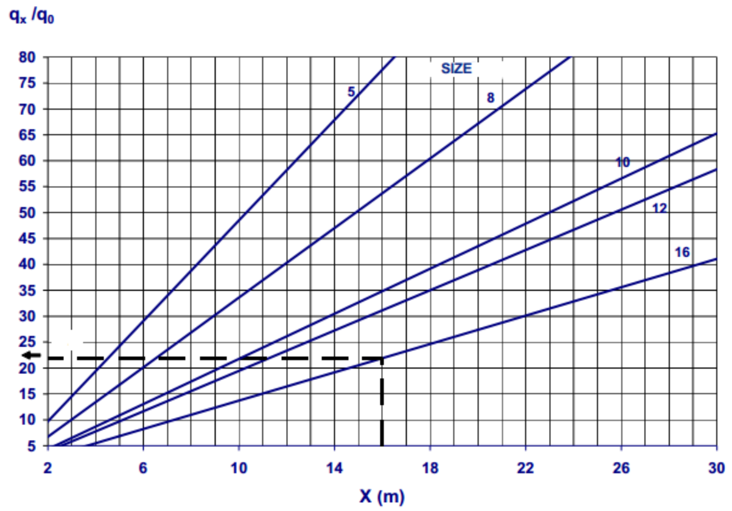


Figure 5. Induction rate

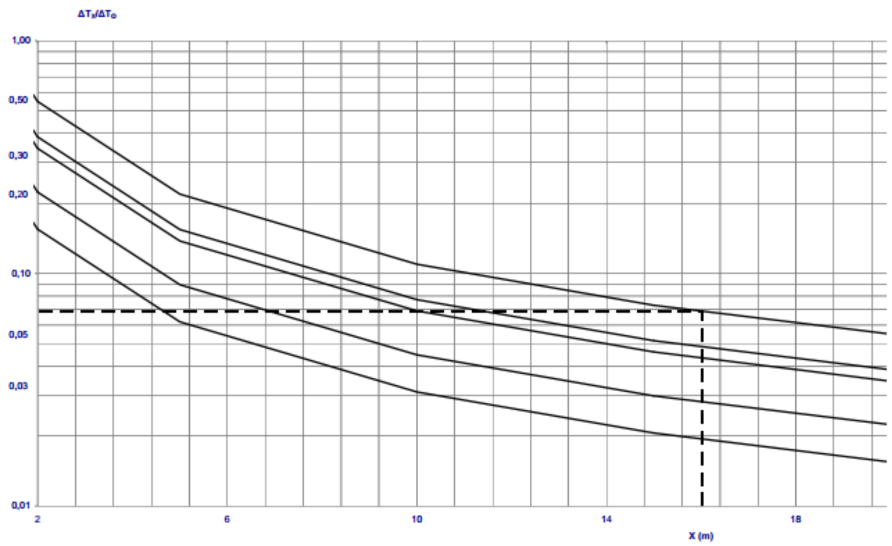


Figure 6. Ratio between temperature differences